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BENJAMIN F. ISHERWOOD, ENGINEER AND SCIENTIST.

A TRIBUTE OF RESPECT BY WALTER M. MCFARLAND, FORMERLY CHIEF ENGINEER, U. S. NAVY.

When a great name has become a memory by the death of the owner, there is a certain satisfaction to those who have been privileged to know and admire his work, in placing on record their tribute of respect and admiration. Admiral Isherwood is undoubtedly considered the greatest marine engineer in the history of this country, and all of us who had the honor of being colleagues of his by service at the same time in the old Engineer Corps of the Navy, are very proud indeed of that fact. The luster of a famous man illuminates to some extent even his unimportant associates.

I never knew Admiral Isherwood intimately, but met him a number of times, and once for a few days, after his retirement, was detailed to do a little unimportant work for him. This last, however, gave a chance to see more of him in an

intimate way, and I have always been glad that the occasion offered. In what follows I shall not attempt an accurate biographical sketch, but shall rather tell of some of the important events in his career, most of which occurred before my active service, as he was retired five years after I left Annapolis.

The most casual acquaintance with Isherwood's record develops the fact that he was a born experimenter and investigator, but some of his early work was done so long ago and has been so overshadowed by the later events of his career as to have been forgotten.

His first book, entitled "Engineering Precedents," was published about 1856, and was well known to the engineers who are now passing away, but I doubt if many engineers who are now under sixty know much about the book. Some years ago I had occasion to make a rather careful study of the early literature of steam engineering in the last century, and I was amazed to find that this book of Isherwood's was the first one to give a reproduction of indicator cards actually taken from marine engines. Other books treated of the indicator. and gave ideal diagrams drawn from theory, but his was the only one to give the actual diagrams. A few years ago, when. in company with some other engineers, I called upon him, I brought this matter to his attention, and asked if he could remember having ever seen any earlier reproductions, and he said he believed his were the first.

Probably the experiments which gave him the widest notice throughout the engineering world were those which he conducted from November, 1860, to February, 1861, on the machinery of the paddle-wheel steamer *Michigan,* at Erie, Pa., to determine the relative economy of using steam with different measures of expansion. At this day, when high pressures are so common, and the question of using steam expansively is as much a matter of course as the high pressures, it is perhaps a little difficult for the younger members of the profession to realize that there ever was a time when it was the

subject of discussion. It must be remembered, however, that in 1860 steam pressures were only from twenty to thirty pounds above the atmosphere. This was also before the days when steam jacketing was common, and the engines were slow running. Isherwood's study of the subject had led him to believe that under these circumstances a moderate ratio of expansion only was desirable, and that if the attempt was made to use higher ratios, instead of producing an increase of economy, as was claimed by many, it would result in an actual diminution. The experiments on the *Michigan* proved conclusively that with such low pressures, and the other conditions named, the economical ratio of expansion was quite low. In fact, it was just about two. It was not greatly different for a moderate range on each side of this figure, and when it is remembered that greater ratios of expansion meant a larger and heavier engine, it is easy to see that all the arguments favored very moderate expansion.

In 1861, Isherwood was made Engineer-in-Chief of the Navy, and a little later the Bureau of Steam Engineering was created, and he was made its Chief, a position which he held for two terms,—until 1869. His service covered the entire period of the Civil War, and his work during that period as designer, administrator and experimentalist deserves the highest praise. Most men would have considered they had done their full duty in looking out for the designing and building of machinery and of the administration of the Bureau, without carrying on extensive experiments. He realized, however, that here was a remarkable opportunity for gaining a vast amount of valuable information, while breaking in the crews of the new vessels, and thereby giving them a higher efficiency from the very start when they got into active service. These experiments covered the expansion of steam with varying points of cut-off, the effect of superheat of varying degrees, forced draft, and numerous other engineering questions. I found only recently that a gentleman who contributed a paper on superheat to one of the Engineering Societies, which dealt

with the question somewhat historically, was apparently unaware of the splendid work which Isherwood did back in the early 60s. Just here it may be appropriate to remark that the famous *Wampanoag* and her equally successful but not so well known sister, the *Ammonoosuc,* were both fitted with superheat, obtained in separately-fired boilers.

The controversy about expansion had not been ended by the *Michigan* experiments, and a prominent patent attorney in New York, possessed of considerable influence, still stood out for the benefits of high rates of expansion with low steam pressures. He had influence enough to have engines of his design fitted to the U. S. S. *Pensacola,* where they proved an unmitigated failure. Even this was not enough, and at a later period he was permitted to design engines for the paddle-wheel steamer *Algonquin.* She was tried out in competition with a sister vessel called the *Winooski,* with engines of Isherwood's design, in February of 1866. Here, again, the machinery with excessive expansion proved a failure. It would seem that two failures ought to have been enough, but permission had been given for the installation of these engines with high expansion on the steamer *Idaho,* which was tested out in May, 1866. Some idea of the failure may be obtained from the fact that the average speed for twenty-four hours was 8.27 knots, as against the fifteen knots required by the contract.

The mention of the *Wampanoag* warrants more extended comment upon that famous vessel, which in her day was the most remarkable one in the world, and, when everything is taken into account, may possibly be considered the most remarkable ship for speed that has ever been produced. She was one of a class of vessels designed to hunt down and destroy the Confederate vessels like the *Alabama* which had been preying upon American commerce, and it was therefore intended that they should have a speed much higher than any other vessel afloat. Isherwood was mainly responsible for the entire design of the vessel, because he suggested the details of

form for the hull. It is not intended to detract from the credit due Naval Constructor Delano, who designed and built the hull, but merely to give Isherwood full credit. In those days the engines ran very slowly and, in order to get the necessary speed of the propeller, gearing was used, but for multiplication instead of reduction, as is now the case with steam turbines. The trial of the *Wampanoag* lasted for 37½ hours at sea, during the last twelve of which there was a gale. The average speed for this whole period was 16.7 knots. For several periods of two or more consecutive hours the speed exceeded 17 knots, going as high as 17.4 average for two hours. If it be remembered that the highest speed of crack merchant steamers at that time was not above 14 knots, it can be realized what a wonderful performance this was. It would be supposed that there must have been the greatest rejoicing in the Navy to know that we had demonstrated the possession of the fastest ship by far in the world, with the expectation (which was subsequently fulfilled) that several sister ships would prove as successful. Curiously enough, this was not so, and instead of the *Wampanoag* being the pride of the Service, she was most heartily damned, for a variety of reasons. Among these such gems of engineering wisdom were displayed as that her grate surface was 1.53 times the midship section; also that one of the smoke-pipes interfered with setting the main-sail, and that the four-bladed screw nullified the use of canvas. Those who are interested in studying the story of the *Wampanoag* in detail will find it splendidly set out in Captain Bennett's book entitled "The Steam Navy of the United States," and I shall not go into the matter further. I may remark, however, that this indifference to the merit of high speed had not died out twenty-five years later, when the *Minneapolis* and *Columbia* repeated the remarkable success of the *Wampanoag,* and again captured the record for the fatest war vessels afloat. I do not remember that they were condemned as absolutely worthless. as was the case with the *Wampanoag,* but I do know that they were not favorites. Let

us hope that the objection to high speed has now passed away, so that when our Battlecruiser comes out, and beats the world's records by several knots, as we all hope will be the case, she will not be condemned because she has not as heavy armor as one of the slower dreadnaughts, and does not carry as many big guns. A little story which I heard from his own lips illustrates the practical side of Isherwood's engineering ability, and will, I think, be appreciated. A new vessel which had recently been completed at the New York Navy Yard was to have a dock trial, and immediately on its completion proceed to her station on the blockade. When steam was turned on, the engine would not move. Examination was made of everything which might apparently have caused the trouble through some maladjustment, and apparently everything was all right. Isherwood wired them to keep steam at full pressure on the pistons, and jack the engines around by hand. This was done, and he went over to New York himself to investigate. He had suspected that the trouble was due to swelling of the lignum-vitae strips in the stern bearing, and as it was not practicable to dock the ship and remove the bearing, he concluded to grind it out until the engine could work sufficiently freely to let her go to sea, when regular service would speedily remedy the trouble. Continuation of the prescription which he had given eventually enabled the engine to make some 25 or 30 revolutions per minute under steam, and she was then sent to sea.

The foregoing account gives some idea of the practical side of Isherwood's nature, but, in fact, he was not only a great scientist and a great engineer, but a man of affairs who could take broad general views and look beyond the immediately attractive features of his work as an engineer, to the far more important feature of the general result which his work as an engineer was expected to assist. He was criticised not only by the advocates of extreme expansion for failure to carry out their ideas, but also by some well-meaning and competent engineers, who were engineers only, for building machinery which they considered inordinately heavy. I once

heard him discuss this matter, and this discussion shows how admirably he was fitted for his position as Engineer-in-Chief during the stress of a great war.

Under the stress of war conditions, the Navy was enormously expanded, with the result that the demand for engineers was very much greater than could be supplied by men of adequate training and experience. In fact, it was necessary to accept as engineers men who had any knowledge of machinery at all who were willing to go to the front and risk their lives. To place in the hands of these inexperienced men complicated machinery, designed to withstand only the stresses which would come upon it due to regular working in the hands of competent and experienced men would have been to risk disaster. Isherwood realized that the supreme demand upon the machinery was that it should always be ready for service, and should never break down. To secure this meant building the machinery much stronger than would ordinarily be the case, and, consequently, much heavier. His foresight, and his indifference to the criticism of men of less breadth of view, resulted in machinery which did not break down, and which enabled vessels to take part in the very successful naval engagements, and, above all, to maintain, without failure, the blockade of the Southern coasts, which is now generally admitted to have been one of the most important factors in the termination of the war. In other words, Isherwood remembered that, while he was an engineer, he was also—and in a more important sense—the head of the Engineering Division of a great military arm of the Government, and that his first duty was to insure military success, even at the risk of criticism from hypercritical devotees of engineering pure and simple. That his course was the correct one is shown not only by the fact that the vessels with his machinery did not break down, but still more by the cases which have already been cited, of the breakdown and failures of the machinery designed by devotees of what they considered the High Science of Engineering.

During his long and busy term as Chief of Bureau Isherwood made a great many enemies in and out of the Service, as was almost inevitable with a man who had so much to do, and who was naturally aggressive and impatient of petty fault-finding. As a result, when his second term expired, he was exiled to the Pacific Coast, as Chief Engineer of the Mare Island Navy Yard, where his enemies doubtless thought there would be nothing for him to do but gnash his teeth in impotent rage. On the contrary, this gave him an opportunity for some of the finest experimental work he ever did,—namely, his famous propeller experiments. His fame was so great that, when these experiments were published in the Annual Report of the Navy Department, they were copied all over the world. So far as I know they were the first experiments where the actual thrust of the screw was accurately measured. These experiments were a mine of information for people interested in screw designing, and I well remember, more than twenty years after they were conducted, reading a series of articles on the screw propeller by an eminent English engineer, who stated that practically all of his accurate data had been derived from these Isherwood experiments at Mare Island.

As showing Admiral Isherwood's methods of work, it will perhaps be interesting to give a little information which I learned from the late Mr. William R. Eckart, a prominent consulting engineer on the Pacific Coast, who was Superintendent of Machinery under Admiral Isherwood when he was Chief Engineer at the Mare Island Yard. It seems that Isherwood had a great many matters under experimental determination, and in his library was a large oblong table on which were placed at different points the papers relating to each particular subject. Mr. Eckart stated that Isherwood's method of relaxation during an evening was to shift from one of these subjects to another when the first became a little tiresome. This seems to be characteristic of some great men, as a somewhat similar course was followed by the late Mr. George Westinghouse, except that with him it was the verbal discus-

sion of subjects with his subordinates, where he would shift from the work of one company to another.

After the termination of his duty at Mare Island Isherwood was ordered East, and during the rest of his career on the active list was President of the Experimental Board, which conducted a great many tests of all kinds, some of which were of very great value. Among these may be mentioned the experiments on the machinery of some Herreshoff yachts, where he investigated the effect of variable rates of expansion with different steam pressures, and also the effect on economy of varying the cut-off in the low-pressure cylinder of a compound engine. All of these tests were fully reported in the annual reports of the Bureau of Steam Engineering.

I have had occasion repeatedly to speak with admiration of Admiral Isherwood's reports on his experiments, which I regard as models worthy of copying by everyone who does such work. He always described completely every part of the apparatus, even if it had nothing particular to do with the experiment which he had in hand. The result of this is that as he described hull, machinery and boilers of every ship, we can go back to his experiments sure of getting any information that we need about the vessel. He also recorded all the data with respect to the performance, whether it was immediately useful to him or not, realizing that experiments are always costly, and that the opportunity should never be lost to get all the data possible, so that if, later on, it is desired to investigate other features, the data will be there. He was a remarkably clear thinker, and expressed himself in clear and elegant language, which it is always a great pleasure to read.

I neglected to say, when speaking of the enormous amount of experimental work done during the Civil War, that these experiments were published by subscription, and entitled "Experimental Researches in Engineering." In the preface he remarks, "The manuscript of these pages was offered to several publishers who declined to print it because they considered such a work too unremunerative. The Navy Department had no funds that could be appropriated to that purpose,

but granted permission for its publication by several eminent engineers who defrayed the cost, in order to have the information in a convenient form."

It is not a matter of official record, but it was well known to the men of Civil War times, and has been stated to me more than once by some of them, that, besides his engineering work, Isherwood was one of the most trusted advisers of Secretary Gideon Welles, and rendered most important and valuable service in the discussion and decision upon matters of policy, and in the presentation of many matters to the general public. It is easy for those of us who know his remarkable mental ability, and his skill as a writer, to understand that he could be of very great assistance to the famous Naval Secretary of the Civil War.

He lived to be very old,—more than thirty years after his retirement. Until the last few years he continued his intellectual labors, and the pages of our JOURNAL during the first ten years have several contributions from his pen. His intellectual faculties were unimpaired until almost the very last days of his life. On the occasion already referred to, when the writer and several other engineers called upon him, we were all astounded at his vigorous mind and his intensely interesting conversation. He had then failed physically, and was no longer the erect, handsome man that he continued to be until over seventy, but was considerably bent, and quite feeble. From the moment that the conversation began, however, his eye lighted up, his voice was firm and resonant, and it was almost impossible to realize that we were talking to a man who was almost ninety years of age.

I realize how very far short I have come of giving an adequate idea of this great man, who was in his day one of the shining lights in the Navy, and of whom we are all justly very proud. I have tried, however, to tell something of his work and a little of the man, with the hope that to those who knew him it will affectionately recall a familiar figure, and to the younger men, who did not, it will give a more vivid idea of his personality.

THE VOLATILE MATTER OF COAL.

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The term "Volatile Matter," which always appears in the report of a bituminous coal analysis, lacks the complete definiteness which we associate with the other items of the analysis, namely, "Ash," "Moisture," "Fixed Carbon," etc.

The object of this paper is to explain more fully the meaning of the expression "volatile matter of coal," the methods used commercially in the treatment of volatile matter, the constituents into which this treatment divides it, and the properties possessed by the products obtained from volatile matter which are of special interest to the Navy.

The volatile matter of a coal is of the same quality and quantity whether the dry coal is carbonized in a platinum crucible, a gas retort or a by-product coke oven; provided the methods of drawing off the volatile matter, the degree of heat applied, and the ratio of the area of hot surface exposed to the charge of coal remain the same. To accomplish the purpose of this paper it is better to consider large volumes of volatile matter such as are produced in by-product coke operations which have specialized in the treatment of the volatile matter of coal.

TREATMENT OF VOLATILE MATTER.

Immediately after dumping the coal into a by-product oven the oven is sealed so as to exclude air and then heated until no more gases or vapors are given off. An oven taking a charge of 10 tons of coal will require 18 hours of heating to drive off completely all the volatile matter. The substance remaining in the oven after all the volatile matter has been driven off is called "Coke," and technically it is the "Fixed Carbon," "Ash," and the major portion of the "Sulphur" of the original coal. By weight it is about seventy per cent. of the

original coal. The coke at a bright red heat is removed from the oven, quenched with water, dried and sorted.

The volatile matter from the coal contains a considerable quantity of heavy tar varpors, which condense from the gas at relatively high temperatures and collect in the gas mains leading to the condensers. The purpose of the condensers is to remove by cooling the tarry vapors in the volatile matter and to cool the gas for subsequent washings with water and oil. Two general types of condensers are in use, the first is a tubular condenser, using cooling water which circulates through the tubes, but does not come in contact with the gas; the second type, which is by far the most efficient, consists simply of 3 or 4 square steel tanks filled with grids over which water and light tar are sprayed. The water and light tar are kept constantly in circulation by means of centrifugal pumps. The liquids are run continuously through an outside cooling coil in order to get rid of the heat taken from the gas. The gas flows in the opposite direction to the liquid circulation; therefore, the hottest gas meets the hottest liquid and the coolest gas is finally sprayed with the coolest liquid. A small stream of water is kept constantly running into the final condensing tank, so as to gradually renew the water in the cooling system, which otherwise would become so saturated with salts of ammonia that it would become solid at ordinary temperatures. The operation, when skilfully performed, removes the tar from the volatile matter at temperatures at which the tar is quite warm, thus preventing the latter from absorbing the light oil vapors in the volatile matter and from becoming chilled and clogging up the apparatus.

The substances condensed and absorbed from the volatile matter in this treatment consist of salts of ammonia—such as sulphate, carbonate, sulphide, etc.—and the coal tar. The heavy tar condensate from the gas mains is added to the condenser liquids and all are allowed to settle. The tar, being heavier than the water solution of ammonia salts, settles to the bottom and is pumped to the tar storage.

Unless some means were provided to draw the volatile mat-

ter away from the ovens excessive pressure would be developed in the ovens, and practically all of the volatile matter would be lost by leaking through the walls of the ovens. Even though it might be possible to build ovens which would be sufficiently tight to prevent leakage, the pressure, caused by the evolving gases and the resistance of the condensing and washing apparatus, would be so great that serious decomposition of the valuable constituents of the volatile matter would take place. Therefore, the gases are lead through an exhauster after passing through the condensers. The exhauster reduces the pressure of the gases in the ovens to about zero, thereby drawing the gases through the tar condensers and thence forcing them through the remaining apparatus. In order to make the power consumption by the exhauster as small as possible, modern practice utilizes methods of washing and scrubbing which offer the least resistance to the flow of the gases and vapors.

The exhauster forces the gas through a water scrubber which frees the gas from ammonia, and then forces it through an oil washer which removes all the light oil vapors in the gas. Many different forms of apparatus are used for these operations, but the main object is to bring the gas into intimate contact with the washing liquids in such a manner that the gas which is richest in ammonia and light oils first comes in contact with water having the most ammonia in solution, and absorbing oil containing the highest percentage of light oil. As the gas passes through the scrubber or washer it is successively washed by the liquids with lower percentages of absorbed constituents, the washing liquid traveling in the opposite direction to the flow of gas; and, finally, the gas, free of ammonia, is washed with fresh water and the gas from which all light oils have been absorbed is washed with fresh absorbing oil.

The volatile matter consists of fixed gases after it has passed through the oil washer and no more products are extracted from it. We have divided the volatile matter of coal by the

foregoing operations into four separate products: tar, ammonia liquor, light oils and residual gas. Not taking into account the variable quantity of water vapor always occurring in volatile matter, the composition by volume of the volatile matter as it leaves the oven, is: tar about 1.5 per cent.; ammonia 1.2 per cent.; light oils 1.0 per cent.; residual gas 96.3 per cent. By weight, the average American coal contains 30.0 per cent. of volatile matter, of which 5. per cent. is tar, 0.25 per cent. ammonia, 1.10 per cent. light oils and 23.65 per cent. fixed gases at ordinary temperatures.

COAL TAR.

Coal tar is a black, heavy liquid of specific gravity about 1.10. At temperatures of 40 to 50 degrees F. it is so viscous that it is pumped with difficulty, but can be easily handled at temperatures of 70 to 80 degrees F. It is but little inferior to petroleum oil as a liquid fuel, and is to a considerable extent replacing fuel oil in the open-hearth furnaces of steel works. When used as liquid fuel it should be heated to 90 to 125 degrees F. It has a calorific heat value of about 18,000 B.t.u. per pound as against 19,300 B.t.u. per pound for fuel oil. Mr. W. A. Forman, of New York City, in testing out coke-oven tar found that it would evaporate 12.64 pounds of water from and at 212 degrees F. per pound of tar. Volume for volume, it has a higher calorific value than fuel oil, and at prevailing prices is only about one-half the cost of fuel oil. From tar we can obtain for the same amount of money over twice as much heat as we can obtain from fuel oil.

Mr. Ralph Arnold, Geologist of Los Angeles, California at the Pan-American Congress held in Washington, recently, estimated that, at the present rate of consumption, the oil supply of the United States will last only 22 years. New methods for treating petroleum oils, which increase their usefulness, are constantly appearing, and in years to come enormous demands will be made on the supply of petroleum. Judging from the consumption of mineral wealth in past years, the present rate of consumption of petroleum oil is probably about one-half of

what the rate of consumption will be ten years from now. If Mr. Arnold's estimate of the future supply of petroleum oil is even approximately correct, it is a safe conclusion to say that petroleum oil will be too expensive in 10 years to use for fuel in the Navy.

Only about one-third of the coal used for making coke in the United States is carbonized so that the tar and the other valuable by-products are saved. Two-thirds of the coal is coked in bee-hive ovens which are not fitted to treat the volatile matter. From the coal coked in by-product ovens we are obtaining approximately 125,000,000 gallons of tar per year. If we discarded the wasteful bee-hive oven and coked all of our coal in by-product ovens, our yearly production of coal tar would be close to 400,000,000 gallons. When we consider that the available supply of bituminous coal is practically unlimited, a conservative estimate being a supply lasting hundreds of years, it seems that fuel oil, the supply of which is limited, is receiving too much attention and that a great deal of the attention of the commercial world could be better concentrated on coal tar. Coal tar has successfully replaced fuel oil for generating steam, for heating in steel mills and as fuel for Diesel engines without the assistance of any organized scientific work as to the best conditions under which it should be used. The general use of coal tar to replace fuel oil would have the tendency to stabilize its market, prolong the life of petroleum oil and be an incentive to extend the use of by-product ovens. In other words, instead of consuming our surplus petroleum oil we would be constantly building up our surplus liquid fuels by encouraging the saving of this constituent of the volatile matter in the coking of coals.

American practice of coal-tar distillation usually separates the coal tar into the following products in approximately the following proportions:

Water.....	2.0 per cent.	Creosote oils.....	30.0 per cent.
Light oils.....	1.5 per cent.	Soft pitch.....	66.5 per cent.

The light oils from the tar are composed of benzene, toluene, and xylene. They constitute only a small percentage of the

total light oils recovered from the volatile matter. Coal tar is not considered as a source of light oils in the United States.

Commercial creosote oil as produced in the United States is made up of carbolic acid, napthalene and creosote oil. It is used in immense quantities for preserving timber and in making preservative stains.

Soft pitch or "Prepared Tar" is used as a binder in road construction, in preparing roofing felt, in black metal paints, etc.

AMMONIA.

The liquor coming from the ammonia scrubber is a foul smelling liquid containing ammonia salts in the form of sulphide, carbonate, cyanide and sulpho-cyanide. The solution of ammonia salts, condensed from the volatile matter with the tar, is added to this liquor. The combined liquids contain only about one to two per cent. of ammonia, and to make the ammonia saleable the content of ammonia has to be raised by volatilizing the ammonia with steam in a still, and either condensing it as crude concentrated ammonia, containing about 25.0 per cent. ammonia, or else conducting the vapors into sulphuric acid, producing ammonium sulphate. The crude concentrated ammonia is shipped to chemical works and there purified. It is used largely in the production of other chemicals, such as carbonate of soda, sodium cyanide, etc. It is also made into the hydroxide of ammonia, anhydrous ammonia gas for refrigeration, and "sal ammoniac" (ammonium chloride). Ammonium hydroxide is probably one of our most useful chemicals, as it is strongly alkaline and volatile in a boiling solution. The debt of analytical chemistry to ammonia and its compounds is a very heavy one.

Ammonia being composed of two gases, hydrogen and nitrogen, decomposes without smoke and leaves no residue, hence many of the salts of ammonia are used in explosives. Melinite, lyddite and ecrasite contain ammonia in the form of ammonium picrate. The Austrian army uses extensively ammonal, which is a mixture of ammonium nitrate, charcoal, tri-nitrotoliol and aluminum.

LIGHT OILS.

The light oils, after being absorbed from the gas by petroleum oil, are then distilled from the petroleum oil by aid of heat and steam; the vapors of steam and light oil are condensed and the water and light oil separated. The petroleum oil, free from light oil, is cooled and pumped back to the washer. The process is continuous and the absorbing oil is used over and over many times.

The condensed light oils from the still are generally of a light yellow color, specific gravity 0.87, very volatile, about 80 per cent. of them distilling off at 212 degrees F. They are the illuminating bodies contained in illuminating gas, 90 per cent. of the light obtained from coal gas, when it is burned in an open tip, is due to these light oils. The light oils were first sold in this country to gas companies, who added them to their gas to raise its illuminating power. This market for light oils is still a very large one.

Another use for light oils, which has been developed during the last ten years, is as a substitute for gasoline in internal-explosion engines. When used for this purpose the oils should be treated with sulphuric acid and alkali to rid them of tarry matters and then redistilled. This treatment gives a "water white" product with a rather sweet smell, which freezes at about the same temperature as water. During the cold weather ten to twenty per cent. of gasoline should be added to it to keep it from freezing. This "water white" product has many properties which make it an ideal fuel for internal-explosion engines; its vapor, when mixed with air and ignited, has a wide explosive range, varying from 2.5 per cent. to 6.0 per cent.; when generating the same amount of power, it will last 25.0 per cent. longer than gasoline; as its explosive power is considerably greater than that of gasoline the horsepower of the engine will be increased by its use; it has a much more uniform composition than gasoline and it will dissolve the carbon produced by the decomposition of the lubricating oil in the gas-engine cylinders.

The volatile matter from one ton of coal, when treated for light-oil recovery, will yield 3 gallons per ton of coal. Ninety-four per cent. of the total light oils in the volatile matter are recovered in the oil-washing process, the remaining 6 per cent. being taken out by the tar. Our total production of light oils at the present time is probably close to 25,000,000 gallons per year; when all of our by-product coke-oven plants are equipped with light-oil recovery plants this will be increased to about 40,000,000 gallons per year. If all of the volatile matter from the coking of coals in the United States were treated for light-oil recovery we would produce close to 125,000,000 gallons yearly.

At the present time it is highly remunerative to distil fractionally the light oils, thus separating them into the following oils in approximately the following proportions: Benzol, 70.0 per cent.; toluol, 20.0 per cent.; xylol, naphthalene and solvent naphtha, 10.0 per cent. Benzol is used for the manufacture of picric acid, which is largely employed as an explosive. Toluol is worked up into tri-nitrotoluol, the most famous explosive of modern times. The toluol is by far the most valuable constituent and the demand at present is far greater than the supply. One ingenious method, proposed by the British War Office Committee for increasing the supply of toluol, is to wash the coal gas which is to be used by cities for illuminating purposes with a wash oil containing about 5 per cent. of benzol; the wash oil will take up the toluol from the gas and will give up a proportionate amount of benzol to the gas. In this way the illuminating power of the gas is not affected and the toluol can be recovered from the wash oil by distillation, thus making it available for producing tri-nitrotoluol.

Xylol and solvent naphtha are used extensively by the large rubber manufacturing companies as a solvent for rubber.

RESIDUAL GAS.

The residual gas, which amounts to about 10,000 cubic feet per ton of coal, has a specific gravity of .4 (air=1), scarcely

any illuminating value when burned in an open-tip burner, and a calorific power of about 565 gross B.t.u. per cubic foot. The composition of this gas varies considerably with the percentage of volatile matter in the coal, methods of charging the coal, atmospheric conditions and general supervision. The average coal, under average conditions, will give a residual gas of the following composition:

Carbon dioxide.....	2.0 per cent.	Carbon monoxide...	6.5 per cent.
Ethylene.....	2.1 per cent.	Nitrogen.....	4.4 per cent.
Methane.....	34.6 per cent.	Oxygen.....	0.2 per cent.
Hydrogen.....	50.0 per cent.	Hydrogen sulphide	0.2 per cent.

One-half of this gas is sent back to the oven flues and burned to carbonize the coal, the other half is usually used to generate steam or as fuel for gas engines. During the first half period of coking, the residual gas contains a much higher quantity of methane and a lower quanity of hydrogen. Consequently, the calorific power of the gas from this period is much higher. The tar and light oils are also more abundant in the volatile matter from the first half period of coking.

The relative value of volatile matter as regards the calorific value of the coal may be expressed as follows:

Fixed carbon or coke, = 72.30 per cent. of total heat.

Volatile matter.—Tar.....= 5.90 per cent. of total heat.
Light oils.....= 1.60 per cent. of total heat.
Residual gas.....=20.20 per cent. of total heat.

From the above it is seen that it is quite important that coal should be so fed to the boilers that all of the volatile matter is consumed. With some coals the volatile matter will be driven off in almost half the time required by other coals. It is almost impossible to fire this kind of a coal and not lose a considerable quantity of volatile matter or heat, since it is almost impossible to regulate the dampers so as to burn entirely the volatile matter.

The volatile matter of coal and the great variety of products obtained from it are, at the present time, of much interest to the commercial world. This interest is also shared by the Navy, since the future may hold a much larger field of development for products derived from the volatile matter of coal.

U. S. S. *NEVADA*.

DESCRIPTION AND TRIALS.

By Lieutenant Ormond L. Cox, U. S. Navy, Member.

Battleship No. 36, the *Nevada*, is a sister ship of the *Oklahoma* and was authorized by an Act of Congress approved March 4, 1911. The *Nevada* is a twin-screw vessel fitted with Curtis turbines, with geared cruising turbines, and designed for a speed of 20½ knots at 27,500 tons trial displacement, with the main engines developing 26,500 shaft horsepower. She was built under contract by the Fore River Shipbuilding Co., of Quincy, Mass. The contract was signed on January 22, 1912, the price being $5,895,000.00 exclusive of armor and armament, and the time of construction thirty-six months.

By a supplementary contract signed on July 31, 1914, the contract price was increased $50,000.00 and the time of construction extended five months. The supplementary contract covered the installation of a geared cruising unit on each shaft.

At the same time the contractors changed the design of the main turbines, reduced the number of rows of blades, increased the blade clearances and changed the method of securing the blades.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	575–00	
on load water-line, feet and inches	575–00	
overall, feet and inches	583–00	
Breadth, extreme, on L.W.L., feet and inches	95–02⅛	
molded, feet and inches	94–10½	
Depth, molded, at side to main deck, M. S., feet and inches	44–05½	
Draught to load-water line, feet and inches	28–06	
Mean trial displacement, tons	27,500	
Ratio length to beam		6.038
Coefficient of fineness, block		.6168
midship section		.982
load-water plane		.715

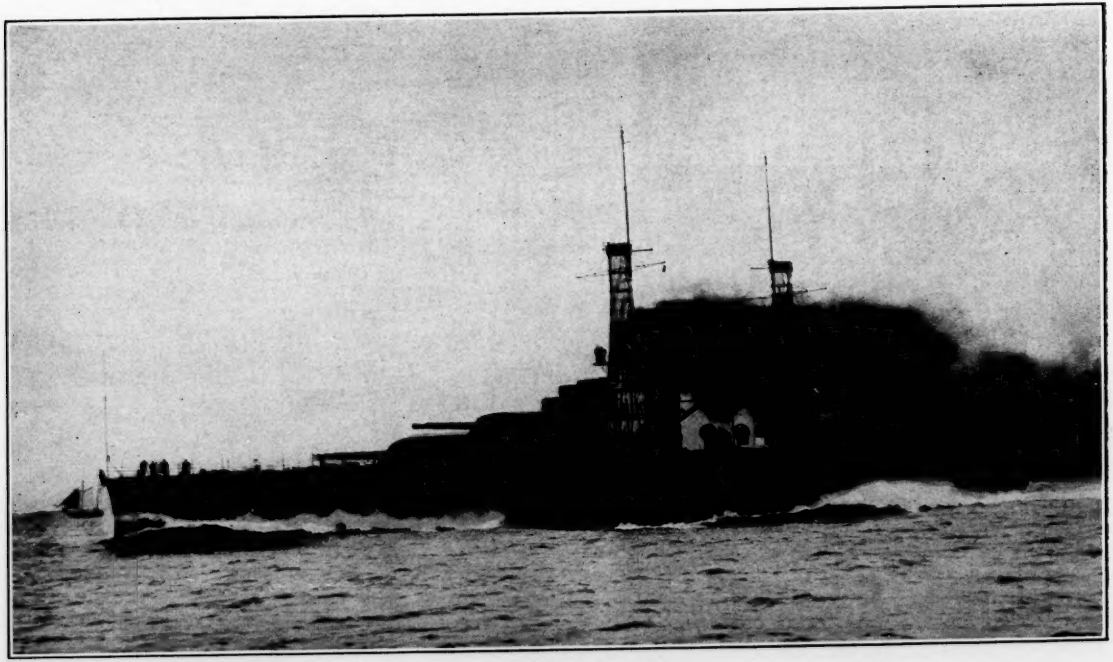

Photo. by LORD.

U. S. S. "NEVADA."

GENERAL DESCRIPTION OF HULL.

The hull is built of steel throughout with all turrets on the center line, one smokepipe, and two cage masts.

Masts.—The masts are one hundred and fourteen feet and one inch from the load-water line to the floor of the spotter's top and are located at frames Nos. 62½ and 90½; on the masts are located the spotter's tops, searchlight platforms—two to each mast, each mounting one searchlight, radio, signal yards, etc.

Conning Platform.—Just forward of the foremast is the conning platform extending, from frame No. 56 to frame No. 59½.

Bridge.—The bridge is just below the conning platform and extends from frame 52 to frame 59½; it has portable extensions at each side, reaching to the vessel's sides. The fire-control and conning towers are located at the forward end of the bridge. The chart house is located on the bridge.

Superstructure Deck.—This deck extends from the after side of No. 2 turret to frame 59½. On it are located two 5-inch guns, one on each side, and four 3-pounder saluting guns, two on each side.

Upper Deck.—The upper deck extends from the stem to frame 85. Forward are the wildcats, an electric-driven winch, and Nos. 1 and 2 turrets. Abaft No. 2 barbette there is a deck house which contains the captain's quarters, consisting of a cabin, stateroom, office and pantry.

Main Deck.—This deck extends from stem to stern and is a weather deck abaft the upper deck. From forward aft are located the following: cleaning-gear compartment in the bow; wardroom officers' water closets, baths, staterooms, messroom, pantry and the officers' galley; part of the five-inch battery, sergeant of marines' stateroom, an electric potato peeler, lamp room, band room and two electric boat cranes at frames Nos. 75–76, port and starboard sides.

Just abaft frame No. 75 is a deck house containing the crew's bakery, butcher shop and galley. Farther aft are Nos.

3 and 4 turrets, an electric winch on each side at frame 96 and an electric capstan amidships at frame 120.

BATTERY.

There are ten 14-inch guns in the main battery, arranged in four turrets, on the center line of the ship. Turrets Nos. 1 and 2 are located on the upper deck, No. 2 being elevated for ahead fire over No. 1. Nos. 3 and 4 are on the main deck aft, No. 3 being elevated for astern fire over No. 4. Turrets Nos. 1 and 4 have three guns each and Nos. 2 and 3 two guns each.

The secondary battery consists of twenty-one 5-inch guns, which are located as follows: seven aft on the second deck, port and starboard, with one at extreme stern on the center line; twelve on the main deck, port and starboard—four in officers' quarters forward, eight in crew's space; and two on the superstructure deck, port and starboard.

There are twenty-one electric-chain hoists for the secondary battery ammunition, each operated by a 3-H.P. General Electric motor.

There are also the following small guns:

Four air-defense guns.
Four 3-pounder guns for saluting.
Two 1-pounder guns for boats.
Two 3-inch field pieces.
Two 0.30-caliber machine guns.

There are four 6.8-meter by 21-inch submerged torpedo tubes.

MAIN PROPELLING MACHINERY.

The main propelling machinery consists of Curtis turbines, arranged on two shafts, in four watertight compartments, as shown on Plate I. In each forward engine room there is a geared cruising turbine, connected to the main shaft by means of a jaw clutch, and one 132-inch-diameter high-pressure

e 96

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Nos.
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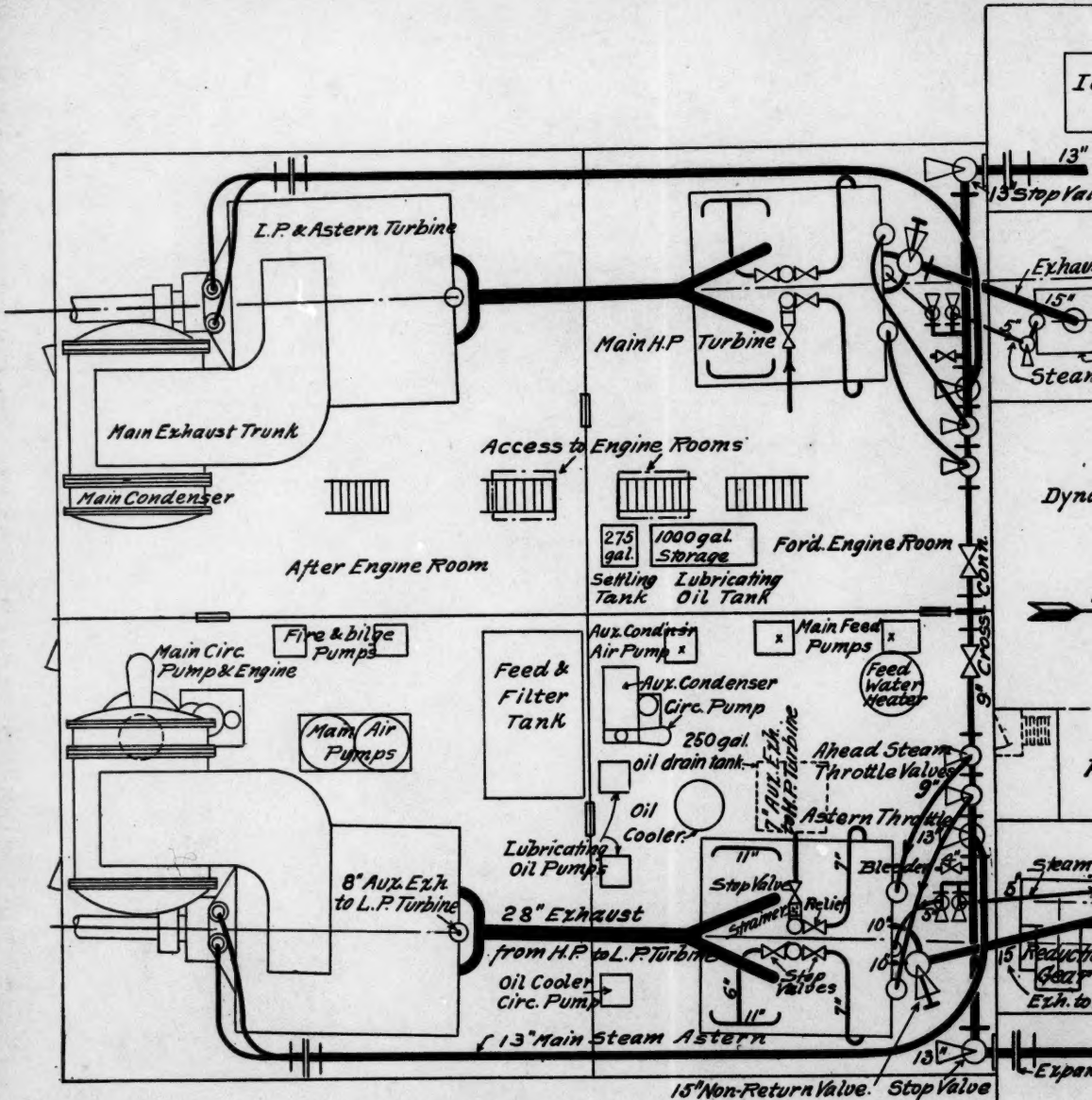


U.S.S. NEVADA Arrangement of Machinery and Main Steam Piping in Engine and Boiler Rooms, and of Fuel Oil Piping in Boiler Rooms. Port and Starboard sides similar except as shown.

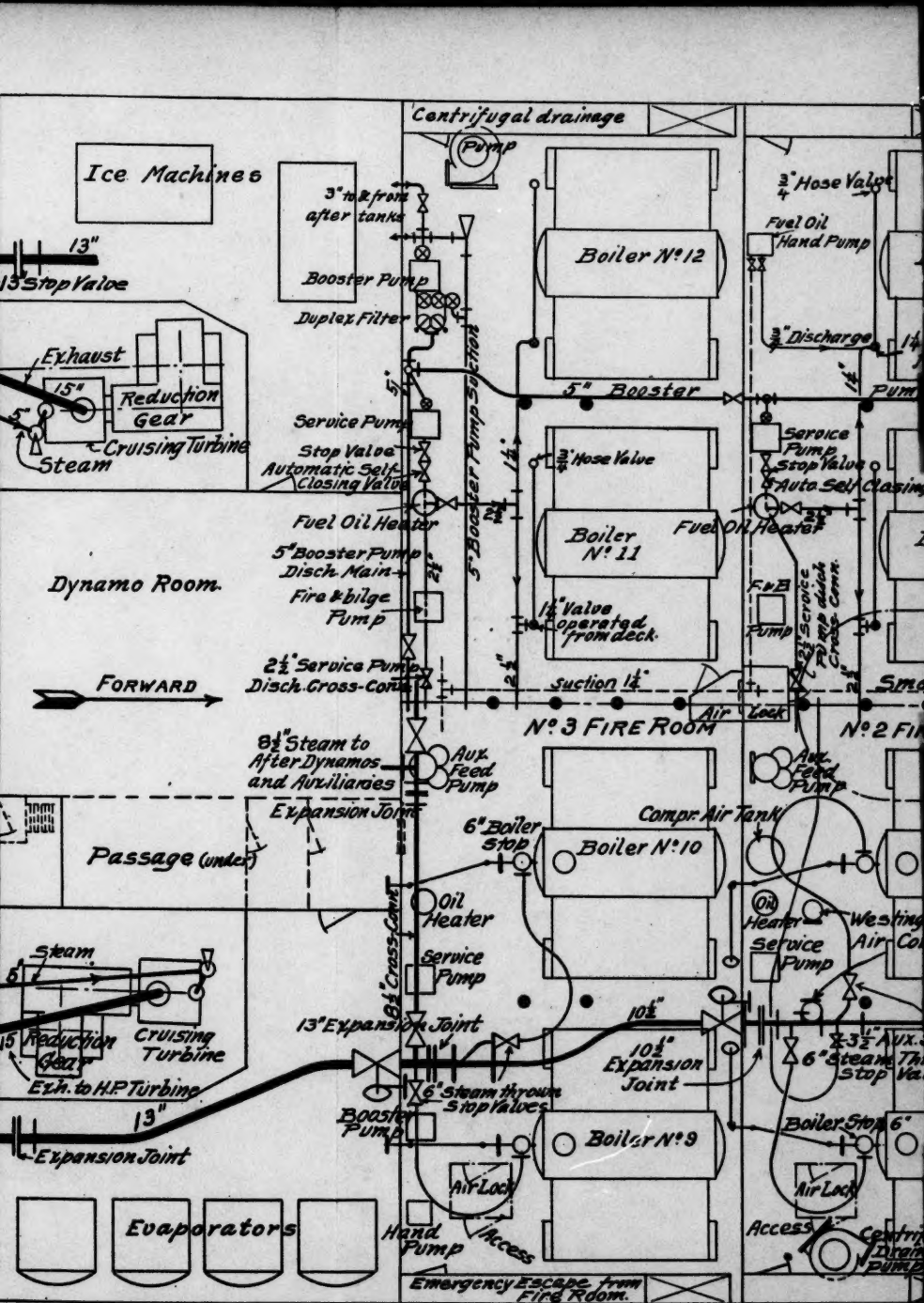

Centrifugal drainage
Pump
Ice Machines
3" to & from after tanks
Booster Pump
Duplex Filter
13"
13" Stop Valve
Exhaust
15"
Reduction Gear
5"
Steam
Cruising Turbine
Service Pump
Stop Valve
Automatic Self Closing Valve
Fuel Oil Heater
5" Booster Pump Disch. Main
Fire & bilge Pump
Dynamo Room.
2½" Service Pump Disch. Cross-Conn.
FORWARD
8½" Steam to After Dynamos and Auxiliaries
Aux. Feed Pump
Expansion Joint
Passage (under)
6" Boiler Stop
Oil Heater
Service Pump
8½" Cross Conn.
13" Expansion Joint
Steam
6
15
Reduction Gear
Cruising Turbine
Exh. to H.P. Turbine
13"
Expansion Joint
Evaporators
Hand Pump
Access
Air Lock
Boiler Stop
Booster Pump
6" Steam through Stop Valves
10½" Expansion Joint
6" Steam Stop
Boiler No. 9
Boiler No. 10
Boiler No. 11
Boiler No. 12
Compr. Air Tank
Oil Heater
Service Pump
Fuel Oil Hand Pump
3" Discharge
5" Booster
3" Hose Valve
Fuel Oil Heater
Fire & B Pump
1½" Valve operated from deck
Suction 14"
Air Lock
No. 3 FIRE ROOM
No. 2 FI
Emergency Escape from Fire Room.

PLATE I.

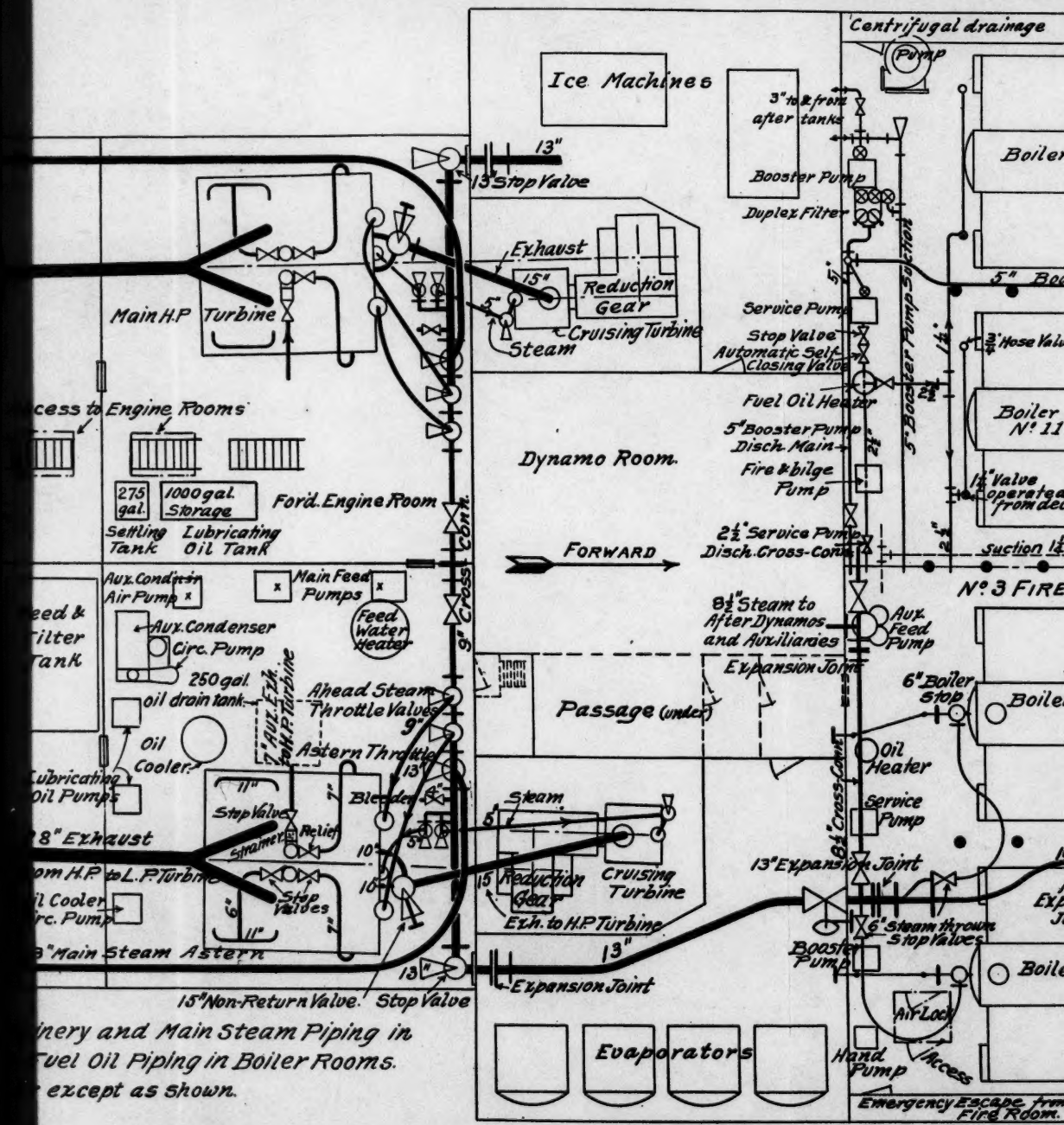
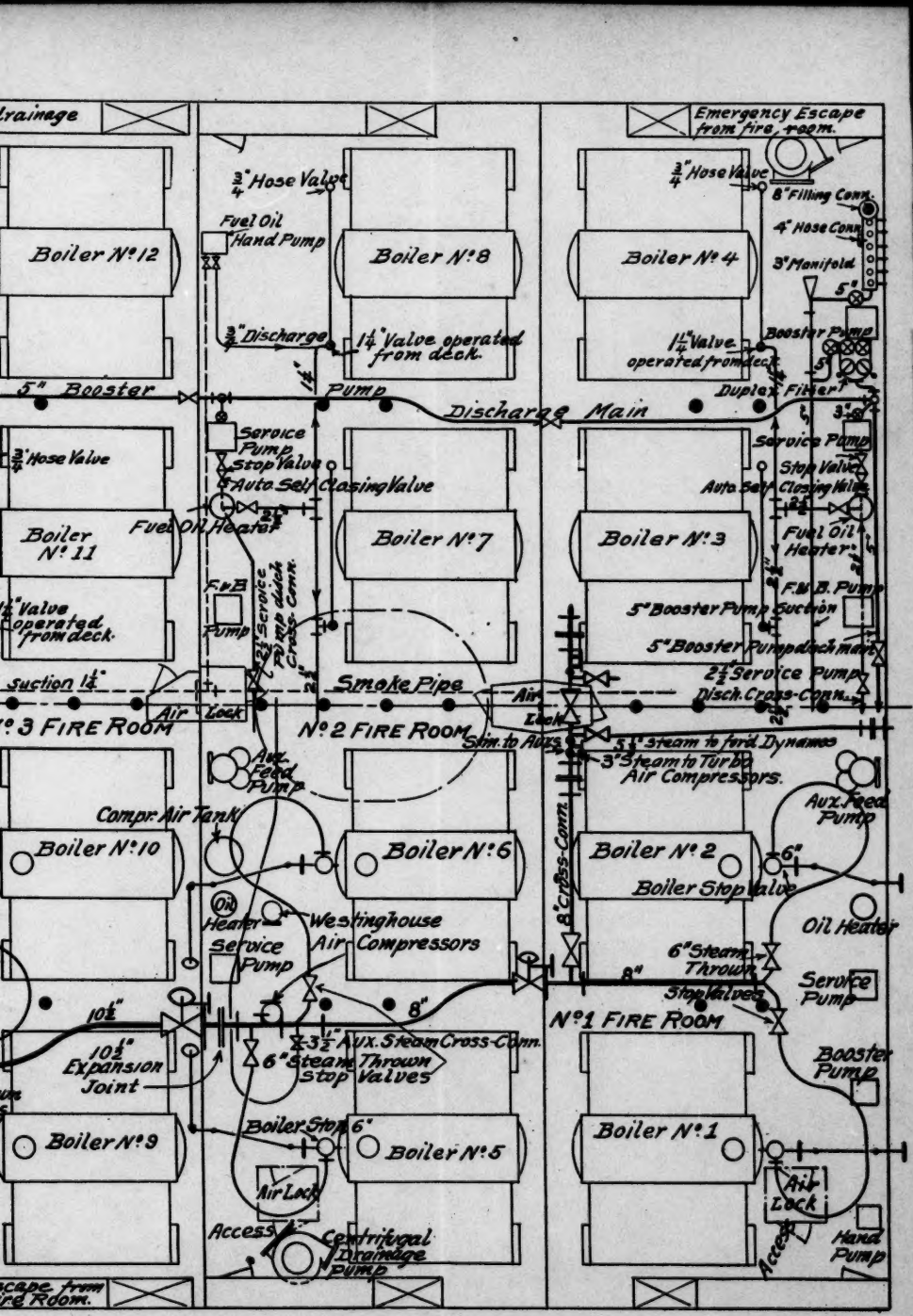

Ice Machines
Centrifugal drainage
Pump
3" to & from after tanks
Booster Pump
Duplex Filter
Boiler
13"
13" Stop Valve
Exhaust
15"
Reduction Gear
5"
Steam
Cruising Turbine
Main H.P Turbine
Service Pump
Stop Valve
Automatic Self Closing Valve
5" Boo
3" Hose Valve
Fuel Oil Heater
5" Booster Pump Disch. Main
Fire & bilge Pump
Boiler No. 11
1½" Valve operated from deck
Dynamo Room.
2½" Service Pump Disch. Cross-Conn.
Suction 1½
FORWARD
No. 3 FIRE
Aux. Condensr Air Pump
Main Feed Pumps
Aux. Condenser
Circ. Pump
Feed Water Heater
8½" Steam to After Dynamos and Auxiliaries
Aux. Feed Pump
Expansion Joint
6" Boiler Stop
Boiler
250 gal. oil drain tank
Ahead Steam Throttle Valves
9"
Passage (under)
Oil Cooler
Astern Throttle
13"
Oil Heater
Lubricating Oil Pumps
11"
Stop Valve
7"
Bleeder
4"
Steam
Service Pump
8" Exhaust
Strainer
Relief
5"
10"
16"
8½" Cross Conn.
13" Expansion Joint
15" Reduction Gear
Cruising Turbine
Exh. to H.P. Turbine
6" Steam thrown Stop Valves
Booster Pump
Boiler
13"
Expansion Joint
15" Non-Return Valve.
Stop Valve
Air Lock
Evaporators
Hand Pump
Access
Emergency Escape from Fire Room.
275 gal.
1000 gal. Storage
Ford. Engine Room
Settling Tank
Lubricating Oil Tank
Engine Rooms
Cross Conn.
inery and Main Steam Piping in
Fuel Oil Piping in Boiler Rooms.
except as Shown.

PLATE I.


Drainage
Emergency Escape from fire room.
¾" Hose Valve
Fuel Oil Hand Pump
Boiler No. 12
Boiler No. 8
Boiler No. 4
¾" Hose Valve
8" Filling Conn.
4" Hose Conn.
3" Manifold
5"
¾" Discharge
1¼" Valve operated from deck.
1¼" Valve operated from deck.
Booster Pump
5" Booster
Pump
Discharge Main
Duplex Filter
Service Pump Stop Valve
6" Auto. Self Closing Valve
Service Pump Stop Valve
Auto. Self Closing Valve
Boiler No. 11
Fuel Oil Heater
Boiler No. 7
Boiler No. 3
Fuel Oil Heater
F. & B. Pump
5" Booster Pump Suction
5" Booster Pump deck main
2½" Service Pump Disch. Cross-Conn.
Suction 1¼"
Air Lock
Smoke Pipe
Air Lock
No. 3 FIRE ROOM
No. 2 FIRE ROOM
Stm. to Aux.
5" Steam to Ford. Dynamos
3" Steam to Turbo Air Compressors.
Aux. Feed Pump
Compr. Air Tank
Boiler No. 10
Boiler No. 6
Boiler No. 2
6"
Boiler Stop Valve
Aux. Feed Pump
Oil Heater
Westinghouse Air Compressors
Service Pump
8" Cross-Conn.
6" Steam Thrown Stop Valves
8"
Oil Heater
Service Pump
10½"
8"
No. 1 FIRE ROOM
3½" Aux. Steam Cross-Conn.
10½" Expansion Joint
6" Steam Thrown Stop Valves
Booster Pump
Boiler No. 9
Boiler Stop 6"
Boiler No. 5
Boiler No. 1
Air Lock
Air Lock
Access
Centrifugal Drainage Pump
Access
Hand Pump
Escape from Fire Room.

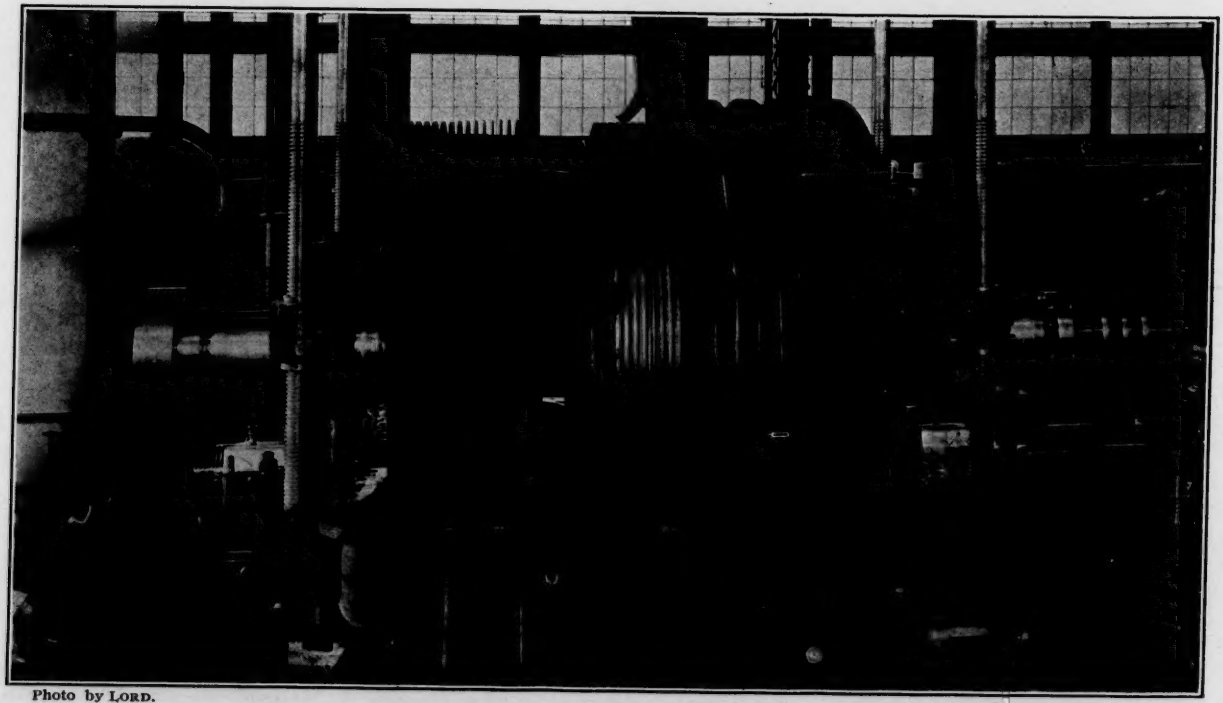


Photo by LORD.

PLATE II.

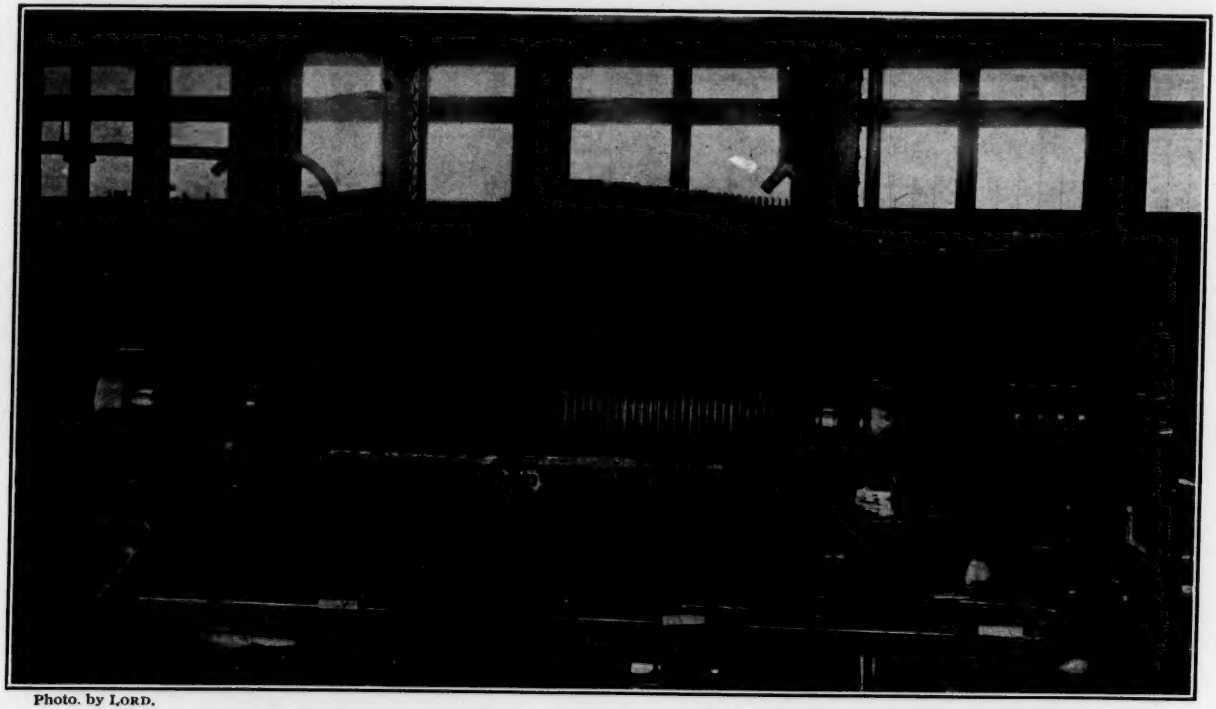


Photo. by LORD.

PLATE III.

turbine. In each after engine room there is one 138-inch-diameter low-pressure turbine and a reversing turbine in a single casing.

The main turbines are designed to develop collectively 26,500 shaft horsepower when making about 222 revolutions per minute. The cruising turbines are designed to develop 1,750 shaft horsepower each at 3,200 revolutions per minute, and are geared down in the ratio of 23.85 to 1. The cruising units may be used at all speeds ahead below 15½ knots and are disconnected for higher speeds and when reversing.

HIGH-PRESSURE TURBINE.

Each high-pressure turbine consists of five wheel stages, a forward and an after drum, as shown on Plate II. The first stage consists of four moving rows, the other wheel stages have two moving rows each. The fifth stage wheel is secured to the forward end of the forward drum. The forward drum has nine single-row stages and the after drum sixteen single-row stages. The two drums are so designed as to partially balance the steam thrust in the high-pressure turbine at all speeds.

LOW-PRESSURE AND REVERSE TURBINES.

The low-pressure and reverse turbines are shown on Plate III. The low-pressure turbine consists of a drum with thirty-eight single-row stages. The reverse turbine consists of one stage with four moving rows of blades, one stage with three moving rows of blades and a drum with eight single-row stages. The second (3-row) stage wheel is integral with the drum.

CASINGS.

Casings of the main turbines are of close-grained cast iron, divided into four parts at the axis on a horizontal and vertical plane. The joints are flanged and are bolted together with fitted bolts.

The interior of the casings has cast-in ribs where required

to support the diaphragms, the ribs being provided with the necessary openings to take the nozzles.

Each casing is provided with four feet for securing the casings to the engine seatings. The after set of low-pressure and the forward set of high-pressure feet are secured by body-bound bolts and cross keys, the other sets having enlarged holes to allow for expansion. The main bearing supports are bolted to the lower half of the casing, the forward one of each turbine being extended to include the thrust bearings.

There are three steam chests in the upper forward section of the high-pressure turbine casing, provided with cast-steel covers. The center chest is for the main steam cruising connection, and the two wing chests for the main steam full-power connections. There is one steam chest in the upper after section of the low-pressure turbine casing, with a cast-steel cover, divided into two parts.

Steam to the high-pressure turbine is supplied through two 9-inch throttle valves and 9-inch pipes, one leading to each of the full-power steam chests. Steam to the cruising chest is supplied through a 5-inch throttle and a 5-inch pipe. For reversing, steam is supplied through a 13-inch throttle and a 13-inch pipe to the after engine room, where the pipe divides into two 9-inch pipes, one pipe leading to each part of the steam chest on the after end of the low-pressure turbine.

For going ahead with main turbines only in use, the course of the steam is as follows: From the main steam pipe in the forward engine room steam is supplied through either one or both 9-inch pipes or the 5-inch cruising connection to the ahead steam chests; thence through the 1st, 2d, 3d, 4th and 5th wheel stages and the forward drum; thence out through four 11-inch receiver pipes to the steam belt on the after end of the high-pressure turbine, forward through the after drum to the exhaust belt of the high-pressure turbine. From there it exhausts through two 20-inch pipes, which combine into one 28-inch pipe, exhaust to the low-pressure turbine. In the after engine rooms this pipe divides into two 20-inch pipes which lead to the steam belt on the ahead end of the low-

pressure turbine. The steam then passes through the low-pressure turbine to the exhaust belt, thence through the exhaust trunk to the condenser.

When the cruising units are in use the course of steam is as follows: From the main steam pipe steam is supplied to the cruising turbine through a 5-inch throttle and 5-inch pipe, through the cruising turbine and out through a 15-inch exhaust pipe, fitted with a non-return valve, into the 1st stage of the high-pressure turbine through two 10-inch pipes; thence through the 2d, 3d, etc., stages as above. Speeds slightly in excess of those given by the steam that will pass through the cruising turbine are obtained by the use of the 5-inch cruising connection to the high-pressure steam chest. It is to be noted that the exhaust from the cruising turbine leads into the 1st stage, and therefore does no work in this stage.

There is a clearance indicator on the forward end of each main turbine for obtaining the fore-and-aft clearances.

There is a 7-inch auxiliary-exhaust connection to the 5th stage of the high-pressure turbine and an 8-inch connection to the steam belt of the low-pressure turbine. There are two 7-inch by-passes from the 1st stage to the 5th stage, one fitted with a relief valve and the other fitted with a stop valve. There is also a 6-inch by-pass from the 5th stage to one of the receiver pipes to the after drum, fitted with a stop valve.

MATERIALS.

The wheels are built up and consist of two steel-plate discs, riveted to a cast-steel spider, the latter being forced on and keyed to the rotor shaft. The wheel rims are of forged steel, except for the 1st stage ahead and astern, which are cast steel.

The drums consist of a cast-steel spider forced on and keyed to the rotor shaft, with heads of steel plate and rims of forged steel.

Diaphragms are of dished-steel plate with cast-steel rim and hub. The central hole through the diaphragm is fitted with a composition bushing to prevent leakage of steam, and is so

designed that it will wear down easily in case rubbing occurs. The recess for the bushing is bored eccentric in the upper half, so that when the rotor is lifted the weight of the diaphragm is taken by the hub instead of on the bushing.

To prevent leakage of steam and air around the shaft where it goes through the casing, carbon packing stuffing boxes are fitted in the usual manner. Provision is made for steam sealing these stuffing boxes.

BLADING.

All blades and distance (packing) pieces are of rolled brass, dovetailed into grooves in the periphery of the wheels and drums. Intermediate blades are made of the same material and secured in the same manner to the holders. Shrouding is of rolled brass.

The principal characteristics of the blading are as follows:

High-Pressure Turbine.

Stage.	Row.	Type.*	Section.*	Length.*	Pitch.*	Pitch diameter. Inches.
	1	M	92522	1.46	.56	
	2	F	82923	1.78	.50	
	3	M	93124	2.09	.56	
1	4	F	83325	2.42	.50	132
	5	M	93627	2.75	.56	
	6	F	84030	3.06	.50	
	7	M	94534	3.38	.56	
	1	M	92923	1 3/8	.56	
2, 3, 4	2	F	83325	1 1/16	.50	132
	3	M	94534	1 3/8	.56	
	1	M	72522	1 1/8	.45	
5	2	F	73630	1 1/8	.45	132
	3	M	74630	1 1/8	.45	

Forward Drum.

6–10	...	N	64520	1 1/8	.44	119 1/4
6–9	...	B	63522	1 1/8	.41	
11–14	...	N	64520	1 1/8	.44	119
10–14	...	B	63522	1 1/8	.41	

*M = Moving blade; F = Fixed blade; N = Nozzle; B = Blade. Section, the first figure gives the width of the blade in eighths of an inch; the next two figures give the entrance angle, and the last two figures the exit angle. Length = Effective length between the packing piece and the shrouding, in inches. Pitch = Pitch of blades or nozzles at packing piece, in inches.

After Drum.

Stage.	Row.	Type.*	Section.*	Length.*	Pitch.*	Pitch Diameter. Inches.
15–20	...	N	64520	2.50	.44	118⅝
15–19	...	B	63522	2.50	.41	
20	...	B	63522	2 1/16	.41	
21–25	...	N	64520	3⅛	.44	118⅝
21–24	...	B	63522	3⅛	.41	
25	...	B	63522	3 7/16	.41	
26–30	...	N	64520	3.75	.44	118⅝
26–30	...	B	63522	3.75	.41	

Low-Pressure Turbine.

1–5	...	N	64520	4⅛	.44	134 13/16
1–4	...	B	63522	4⅛	.41	
5	...	B	63522	4.91	.41	
6–10	...	N	64520	5¼	.44	134⅞
6–10	...	B	63522	5¼	.41	
11–15	...	N	64520	6.00	.44	135⅛
11–14	...	B	63522	6.00	.41	
15	...	B	63522	6 3/16	.41	
16 20	...	N	64520	7.00	.44	136¼
16–19	...	B	63522	7.00	.41	
20	...	B	63522	7¼	.41	
21–25	...	N	65022	8⅛	.50	136⅞
21–24	...	B	63524	8⅛	.43	
25	...	B	63524	9⅛	.43	
26–30	...	N	65022	10⅛	.50	136⅞
26–29	...	B	63524	10⅛	.43	
30	...	B	83828	11⅛	.50	
31–34	...	N	85526	12.00	.56	136½
31–33	...	B	83828	12.00	.50	
34	...	B	83828	12⅜	.50	
35–36	...	N	86032	13.00	.61	137
35	...	B	84534	13.00	.50	
36	...	B	84534	13⅛	.50	
37–38	...	N	86032	13.50	.61	137¼
37–38	...	B	84534	13.50	.50	

Reverse Turbine.

	1	M	92522	1.76	.56	
	2	F	82923	2.13	.50	
	3	M	92923	2.59	.56	
1	4	F	82923	2.95	.50	137.55
	5	M	93124	3.41	.56	
	6	F	83325	3.77	.50	
	7	M	93627	4.20	.56	

Stage.	Row.	Type.*	Section.*	Length.*	Pitch.*	Pitch diameter. Inches.
	1	M	72522	$1\frac{13}{32}$	.45	
	2	F	72522	$1\frac{11}{16}$	.45	
2	3	M	72522	$1\frac{31}{32}$	.45	120
	4	F	72923	$2\frac{1}{4}$	.45	
	5	M	73630	$2\frac{17}{32}$	.45	
			Drum.			
3	...	N	64520	3.50	.44	120
3	...	B	62723	$3\frac{13}{16}$	.45	
4	...	N	64520	$4\frac{1}{16}$	.44	120
4	...	B	62723	$4\frac{1}{8}$	.45	
5	...	N	64520	$4\frac{1}{8}$	.44	120
5	...	B	62723	$4\frac{15}{16}$	.45	
6	...	N	65022	$5\frac{1}{16}$	.50	120
6	...	B	62925	5.50	.45	
7	...	N	65022	5.75	.50	120
7	...	B	62925	$6\frac{1}{16}$	.45	
8	...	N	65022	$6\frac{5}{16}$	.50	120
8	...	B	62925	$6\frac{1}{4}$	.45	
9	...	N	85526	$6\frac{29}{32}$	.56	120
9	...	B	83828	$7\frac{9}{32}$	.50	
10	...	N	86032	$7\frac{1}{8}$	.61	120
10	...	B	84534	8.00	.50	

NOZZLES.

Nozzles for the 1st stage ahead and 1st stage astern are of composition, the openings being accurately machined and filed to template. Nozzles for the 2d to 5th stages ahead and 2d stage astern are of composition with monel-metal division plates. The nozzles for the 1st stage ahead and 1st stage astern are expanding, all others are parallel-wall nozzles.

The cruising (center) steam chest on the high-pressure turbine supplies steam to the seven cruising nozzles, four of which are always open, the other three being controlled by individual valves. Each wing chest supplies steam to eight regular and two overload nozzles, the latter being controlled by individual valves.

The astern steam chest supplies steam to fourteen nozzles, which are always open.

The nozzles for all wheel stages except the 1st stage astern

are 1⅛ inches high; nozzles for the 1st stage astern are 1.3 inches high.

The expansion ratio of the regular and overload nozzles is 1.165; of the cruising nozzles 2.5, and of the astern nozzles 1.9.

The number, location and total throat areas of the nozzles are as follows:

First Stage Ahead.	Throat area. Sq. inches.
16 regular	10.692
4 overload	2.673
7 cruising	3.465
Second Stage Ahead, 106 Nozzles.	
37 in each upper quadrant; 16 in each lower quadrant	48.616
Third Stage Ahead, 120 Nozzles.	
44 in each upper quadrant; 16 in each lower quadrant	55.037
Fourth Stage Ahead, 144 Nozzles.	
52 in each upper quadrant; 20 in each lower quadrant	66.05
Fifth Stage Ahead, 196 Nozzles.	
49 in each upper quadrant; 49 in each lower quadrant	89.897
First Stage Astern.	
14	12.376
Second Stage Astern, 208 Nozzles.	
52 in each upper quadrant; 52 in each lower quadrant	92.59

All of the above nozzles have an exit angle of 20 degrees.

The remaining stages of the turbine are single-row and the nozzles are of rolled brass, built up and secured in the same manner as the blades.

CRUISING TURBINES AND REDUCTION GEARS.

The cruising turbines and reduction gears were manufactured by the General Electric Co., Schenectady, N. Y. The turbines are 3-stage Curtis, designed to develop 1,750 shaft horsepower at 3,200 revolutions per minute, with steam at 250 pounds gage in the steam chest and exhausting against a back pressure of about five pounds gage. All stages are wheel stages, the 1st having two rows of moving blades and the 2d and 3d stages one moving row each. The particulars of the blades are as follows:

Stage.	Row.	Type.	Section.	Length.	Pitch.*	Pitch diameter. Inches
	1	M	82922	0.90	.60	
1	2	F	84026	1.20	.55	38
	3	M	86030	1.68	.552	
2	1	M	82520	0.75	.574	40½
3	1	M	82922	1.00	.585	41

The distance pieces are forged solid with the blades for both moving and intermediate blades. The moving blades for the 1st stage are monel metal, those for the 2d and 3d stages and 1st stage intermediate are nickel bronze. All blades have steel shrouding.

CASINGS.

The casings of the cruising turbines are cast steel. The casing head is divided into two parts on the horizontal center line, the main part of the casing is likewise divided into two parts on the horizontal center line. The turbine wheels are forged steel. The 2d stage diaphragm is cast steel and the 3d stage diaphragm, cast iron.

NOZZLES.

Nozzles for the 1st stage are composition and are carried in the upper section of the casing head. Nozzles for the 2d stage are composition with monel-metal division plates and are carried by the diaphragm. Nozzles for the third stage are cast in the diaphragm and have nickel-steel division plates. There are fifteen nozzles for the 1st stage with a throat area of 3.75 square inches and an expansion ratio of 1.133; forty-eight nozzles for the 2d stage with an area of 12.3 square inches; forty-eight nozzles for the 3d stage with an area of 20.95 square inches.

REDUCTION GEARS.

One cruising turbine and reduction gear is shown assembled on Plate IV; the reduction gear on Plate V; the intermediate gear, shaft and pinion on Plate VI; and the main gear wheel on Plate VII.

* Pitch at pitch diameter.

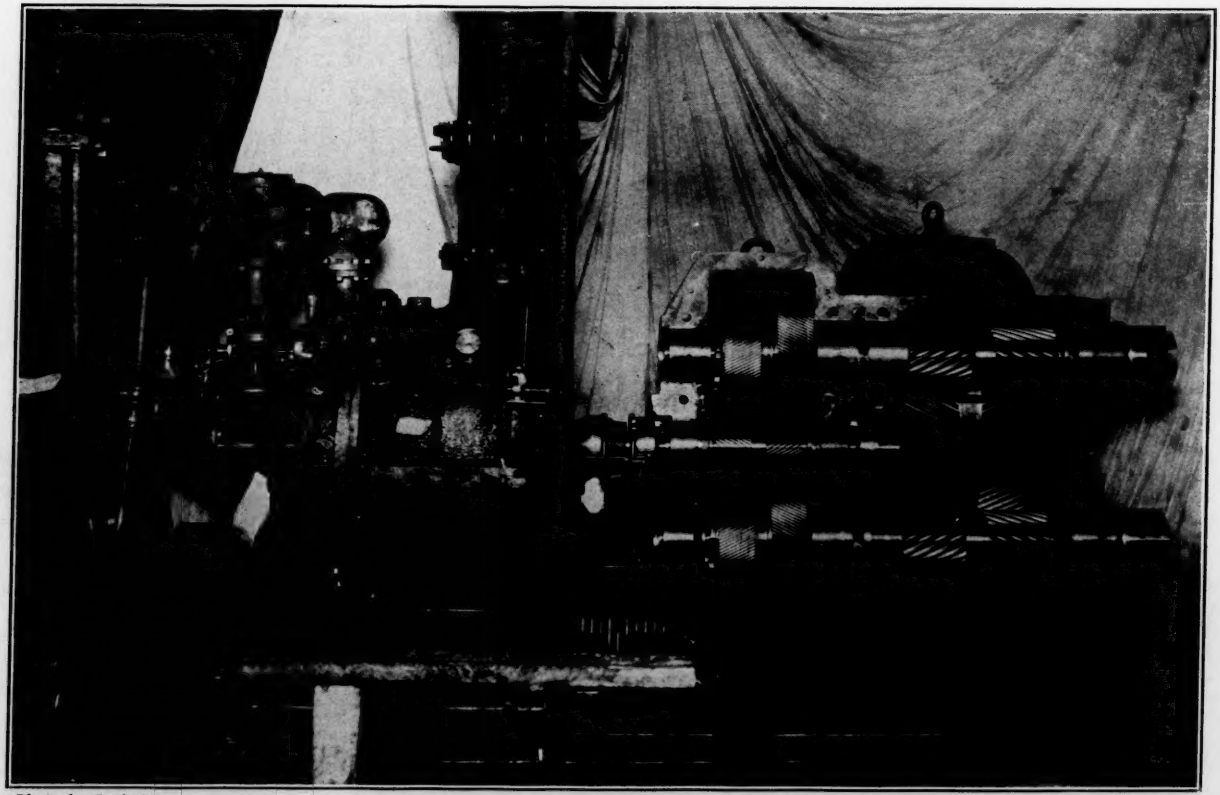


Photo. by G. E. Co.

PLATE IV.—TURBINE AND DOUBLE REDUCTION GEAR FOR U. S. S. "NEVADA."

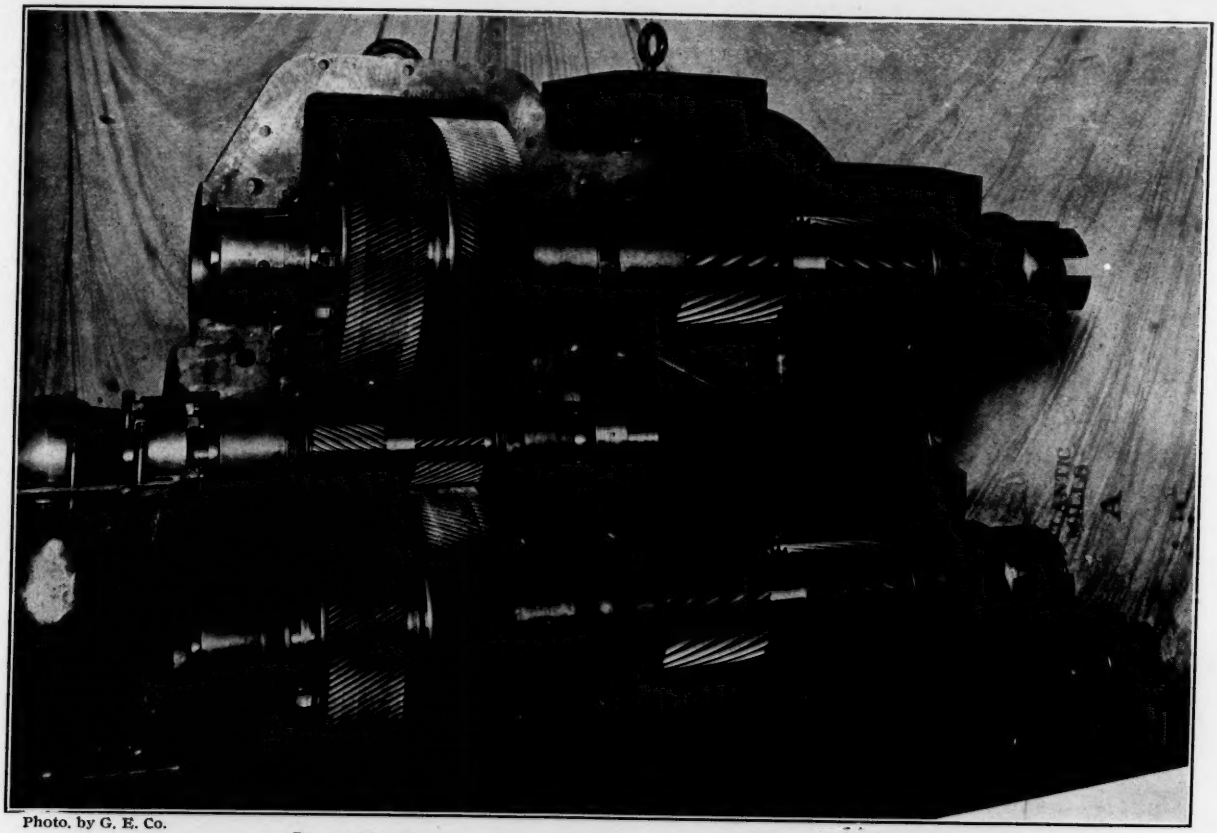


Photo. by G. E. Co.

PLATE V.—DOUBLE REDUCTION GEAR FOR U. S. S. "NEVADA."

PLATE V.—DOUBLE REDUCTION GEAR FOR U. S. S. "NEVADA."

Photo. by G. E. Co.

PLATE VI.—INTERMEDIATE SHAFT DOUBLE REDUCTION GEAR U. S. S. "NEVADA."

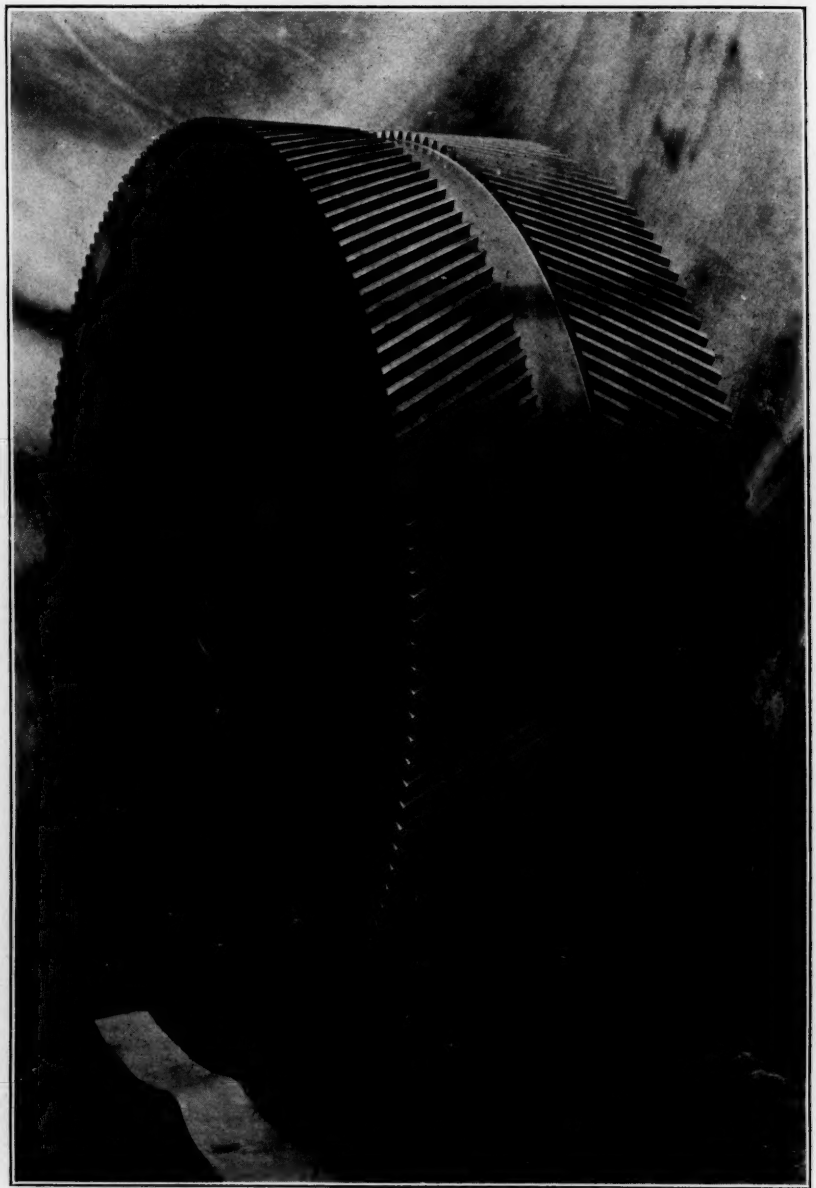


Photo. by G. E. Co.

PLATE VII.—LOW SPEED GEAR FOR DOUBLE REDUCTION GEAR U. S. S. "NEVADA."

Both turbines and gears are identical and turn in the same direction, so that it is necessary to turn the port unit around in order that the shaft will turn outboard when going ahead. This arrangement is shown on Plate I, the main gear shaft being brought out at opposite ends of the gear case for the two conditions. For the port unit this necessitates a section of line shafting between the main gear shaft and the high-pressure thrust shaft.

The gear is the double-reduction type, the main pinion on the turbine shaft drives the gears on the two intermediate shafts and the pinions on the latter drive the main gear wheel. The faces of all pinions and gears are divided into two parts, with the teeth cut at opposite angles in order to balance the thrust. All teeth are cut at an angle of 27 degrees with the axis.

Both the main and intermediate pinions are forged solid on a sleeve which fits around the shaft. The sleeve and shaft are coupled together at one end as shown in Fig. 1, which shows the arrangement for the main pinion.

The intermediate gears are built up, each section consisting of five steel discs, 1.15 inches thick, securely bolted together. The main gear is cast iron with a forged-steel rim in which the teeth are cut.

Particulars of the gears are as follows:

Part.	Pitch diameter. of teeth, ins.	No. of teeth.
Main pinion, ...	6.500	39
Intermediate gear, ...	25.166	151
Intermediate pinion, ...	10.000	25
Main gear, ...	61.600	154

$$\text{Reduction} = \frac{151 \times 154}{39 \times 25} \text{ to } 1\text{; } = 23.85+ \text{ to } 1.$$

SHAFTING AND BEARINGS.

There are two lines of shafting, each consisting of the necessary shafting for the cruising units, a high-pressure thrust and rotor shaft, an intermediate shaft, a low-pressure thrust and

rotor shaft, two sections of line shafting, a stern-tube shaft and a propeller shaft, all of which are supported by the necessary bearings.

All main shafting is hollow-forged steel with disc or special couplings between the various sections. Between the gear shaft and the high-pressure thrust shaft there is a jaw clutch for throwing the cruising units in or out of gear. Between the high-pressure rotor shaft and the intermediate shaft there is a special coupling to allow for expansion of the high and low-pressure shafts.

The high-pressure thrust shaft is coupled to the forward end of the rotor shaft and the low-pressure thrust shaft is forged integral with the rotor shaft.

Each rotor shaft is supported by two main bearings, the high-pressure thrust shaft by a steady bearing, each section of line shafting by two spring bearings, the stern-tube shaft by two stern-tube bearings, and the propeller shaft by two strut bearings. The stern-tube and strut bearings are lignum vitae lined, all other bearings being white-metal lined and the main and thrust bearings are fitted for the circulation of cooling water if required.

The stern-tube shaft is covered with a composition sleeve within the stern tube and is secured to the line shaft by a special disc coupling and to the propeller shaft by the usual sleeve coupling.

The propeller shaft is composition bushed at the strut bearings.

Shaft Data.

H.P. thrust shaft, length, feet and inches	3–00½
diameter, inches	10
axial hole, inches	05
collars, number	3
thickness, inches	01¾
space between, inches	04
O.S. diameter, inches	23
I.S. diameter, inches	10
rotor shaft, length, feet and inches	20–04
diameter, inches	20 to 24⅝
axial hole, inches	14
coupling, inches	22
special, O.S., inches	33⅛

Intermediate shaft, length, feet and inches	3-10⅝
diameter, inches	12
axial hole, inches	06
forward coupling, inches	33½
after coupling, inches	23½
L.P. thrust and rotor shaft, length, feet and inches	26–07
diameter thrust, inches	12
axial hole, inches	06
rotor, inches	22 to 30
hole, inches	15
collars, number	6
thickness, inches	02
space between, inches	04
O.S. diameter, inches	26 1⅛/16
I.S. diameter, inches	12
Forward line shaft, length, feet and inches	27–00
diameter,* inches	14½
axial hole, inches	08
forward coupling, inches	35
after coupling inches	27
thickness, forward coupling, inches	03½
after coupling, inches	03
After line shaft, length, feet and inches	30–06
diameter,* inches	14½
axial hole, inches	08
couplings, inches	27
thickness, couplings, inches	03
Stern-tube shaft, length, feet and inches	49–05½
diameter, inches	14½
axial hole, inches	08
Propeller shaft, length, feet and inches	49–07½
diameter, inches	14½
axial hole, inches	08

Inboard Coupling.

Diameter of sleeve, outside, inches	27
inside, inches	16½
Length of sleeve, inches	10
Thickness of collars, inches	03
Bolts, number each coupling	8
diameter at face of coupling, inches	03

Outboard Coupling.

Length of sleeve, inches	60
Diameter of sleeve, outside, inches	19½

* Diameter at bearings is 14½ inches.

Bearing Data.

Main bearings:	
Number each engine	2
Diameter, high-pressure, inches	20
low-pressure, inches	22
Length, high-pressure, inches	22
low-pressure, inches	28
Steady bearings:	
Number each high-pressure turbine	1
Diameter, inches	10
Length, inches	08
Thrust bearings:	
Shoes, number each high-pressure turbine	3
low-pressure turbine	5
Total effective surface, high-pressure, square inches	780
low-pressure, square inches	1,059
Line-shaft bearings:	
Number each engine	4
Diameter, inches	14¾
Length, inches	20
Stern-tube bearings:	
Number each engine	2
Diameter, inches	16½
Length, inches	60
Strut bearings:	
Number each engine	2
Diameter, inches	16½
Length forward, inches	30
after, inches	60

PROPELLERS.

There is one propeller on each shaft. They are of manganese bronze, 3-bladed, cast solid. The blades are machined true to pitch and the hubs have a taper fit on the shafts, secured by a key and nut.

Propeller Data.

Diameter of propeller, feet and inches	14–03
hub (mean), feet and inches	2–01½
Pitch, feet and inches	12–00
Ratio, pitch to diameter	0.8421
Area, projected, square feet	71.5
helicoidal, square feet	81.5
disc, square feet	159.48
Ratio, projected to disc	0.4483
helicoidal to disc	0.51
Heighth of lower tip of blade above keel, inches	02⅝
Immersion of upper tip of blade, inches	166⅞

MAIN CONDENSING APPARATUS.

Main Condensers.—There is one main condenser in each after engine room. The condensers are pear shaped and have the tubes rolled in the tube sheets at one end and gland packed at the other.

The principal dimensions are as follows:

Thickness of shell (steel), inch	$00\frac{7}{16}$
Length between tube sheets, feet and inches	7–09$\frac{1}{4}$
Thickness of tube sheets, inches	01$\frac{1}{2}$
Tubes, number	7,770
diameter, outside, inch	00$\frac{5}{8}$
thickness, inch	.065
Cooling surface, square feet	9,900
Main exhaust nozzle, feet and inches	7–09 by 6–04
Diameter auxiliary exhaust nozzle, inches	08
bleeder nozzle, inches	04$\frac{1}{2}$
air-pump suction, inches	15
circulating-water inlet and outlet, inches	29

*Main Air Pumps.**—Each main condenser is provided with a Blake vertical, twinplex, single-acting air pump, with steam cylinder 18$\frac{1}{2}$ inches in diameter and wet and dry-air cylinders 40 inches diameter each, with a common stroke of 21 inches. The wet and dry-air suctions are each 10$\frac{1}{2}$ inches in diameter and both take off the common suction stand pipe on the condenser. The wet discharge to feed tank is 10$\frac{1}{2}$ inches in diameter and the dry discharge to atmosphere is 8 inches in diameter. From each wet and dry-air pump suction there is a 7$\frac{1}{2}$-inch cross-connection pipe, with gate valve at pump, for uniting air-pump suctions.

*Main Circulating Pumps and Engines.**—There is one centrifugal circulating pump for each main condenser, driven by a vertical compound engine. The engine is fitted with a self-contained forced-lubrication system. The principal dimensions of the pump and engine are as follows:

*See Table I.

Capacity of pump, gallons per minute (about)	22,500
Diameter of suction nozzles (2), inches	21
discharge nozzle, inches	29
impeller, inches	48
H.P. cylinder, inches	11
L.P. cylinder, inches	22
Stroke, inches	12
Revolutions per minute	235

Feed and Filter Tank.—A combined feed and filter tank of 4,565 gallons capacity is located in each after engine room, forward inboard corner. The filter chamber is in the top of the tank and has a capacity of about 805 gallons. The filter chamber has an inner bottom of loose perforated plates and is divided into three compartments by double vertical division plates. These plates are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following connections:

1 10½-inch main wet-air pump discharge.
1 21-inch stand pipe.
1 4-inch auxiliary air-pump discharge.
1 2½-inch distiller fresh-water pump discharge.
1 7-inch overflow to bilge.
1 3-inch trap-drain discharge.
1 3-inch vapor escape.
3 2-inch vapor escapes, one from each filter chamber.

The stand pipes are cross-connected with an 11½-inch pipe. From the cross-connection pipe there is a 10-inch auxiliary feed-suction main leading to the auxiliary feed pumps in the firerooms, also a 5½-inch independent suction for each main feed pump.

ENGINE-ROOM AUXILIARIES.

Auxiliary Condenser.—In each forward engine room there is a pear-shaped auxiliary condenser, connected through the auxiliary exhaust pipe to all the auxiliary machinery except

dynamos. The tubes are rolled in the tube sheets at one end and gland packed at the other.

Principal dimensions as follows:

Thickness of shell (steel), inch	00¼
Length between tube sheets, feet and inches	4-10
Thickness of tube sheet, inch	01
Tubes, number	513
diameter, outside, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet	405
Diameter of auxiliary exhaust nozzle, inches	08
air-pump suction, inches	04½
heater-drain connection, inches	02
circulating-water inlet and outlet, inches	05

*Auxiliary Air Pumps.**—Each auxiliary condenser is provided with a Blake, vertical, double-acting, single, feather-weight air pump with steam cylinder 6 and water cylinder 10 inches in diameter and a stroke of 8 inches. The suction nozzle is 4½ and the discharge nozzle 4 inches in diameter.

*Auxiliary Circulating Pumps.**—Each auxiliary condenser is provided with a centrifugal circulating pump, driven by a vertical single engine of the following dimensions:

Diameter of suction nozzle, inches	05
discharge nozzle, inches	05
impeller, inches	26
steam cylinder, inches	05
Stroke, inches	05
Revolutions per minute	330

Feed-Water Heaters.—There is a Reilly, vertical, multicoil feed-water heater installed in each forward engine room, inboard side. Each has 319.8 square feet of heating surface, and is used with the main feed pumps only. The heating agent is auxiliary exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve close to each connection to a condenser, and opening towards the condenser.

* See Table I.

TABLE I.

PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Sizes, inches.	Types.	Suction pipes from.	Inch.	Discharge pipes to.	Inch.	Location.
2	Main air.	18½x40x40 21	Twin Vertical single acting.	Condenser (wet)	10½	Feed tank.	10½	In each Aft eng. room.
				do. (dry)	10½	Overflow tank and atmosphere.	8	
				Reserve feed tank	1½			
				Hot suction from outflow tank.	4			
2	Main Circulating.	29" Centrifugal. 48" impeller.	11x22x12 Vertical compound engine.	Sea (1)	23½	Condenser.	29	In each Aft eng. room.
				Drainage.	18½	Oil cooler and water service.	3	
				Dry air pump cooler.	1½			
				Cross-connection.	18½			
4	Main feed.	15½x9½x24	Vertical, piston, double-acting, single.	Feed tank cross-connection.	6½	Main feed discharge line.	5	In each ford engine room.
3	Auxiliary feed.	do.	do.	Feed tank cross-connection.	6½	Main feed line.	5	In each boiler room.
				C.S.R. pipe from ship's sides. (2)	4	Hose connection.	2½	
				Reserve Feed tanks.	2½	Reserve feed tanks.	4	
				Hose connection.	2½	Each boiler in same compartment.	3	
4	Engine room fire & bilge.	12x10x18	do.	Sea.	5	Fire main.	5	In each Aft eng. room.
				C.S.R. drainage.	5	Overboard.	5	
				Hose connection.	1½	Sanitary. (3)	5	
				do. (3)	4	Hose connection.	2½	
						do. (3)	4	
3	Fire room fire & bilge.	do.	do.	Sea.	5	Fire main.	5	In each boiler room.
				C.S.R. drainage.	5	Overboard.	5	
				Hose connection	1½	Sanitary	5	
				do. (4)	4	Hose connection.	2½	
						do. (4)	4	
1	Dynamo condenser circulating.	10" Centrifugal 26" impeller	5x8 Engine.	Sea.	10	Dynamo condenser.	10	Ford dynamo condenser room.
1	do.	9" do.	do.	Sea.	9	do.	9	Aft dynamo condenser room.
2	Dynamo condenser air.	8x18x12	Twin, vertical, single acting.	Condenser (dry)	4	Hotwell (in dyn. room)	4	One in each dyn. condenser room.
				do. (wet)	4	Vapor	3½	
2	Dynamo room hotwell.	4½x5x6	Vertical, piston, double-acting, single.	Hotwell tank.	2½	Aux. feed suct. main.	2	In each dynamo condenser room.
				Dyn. condenser.	2½			
2	Evaporator feed.	4½x5x6	do.	Sea and overboard discharge from distillers.	3	Evaporators through feed heaters.	2½	Evaporator room.
2	Distiller circulating.	12x14x16	do.	Sea.	6½	Distillers.	4	do.
						Fire main.	4½	
2	Distiller fresh water.	4½x5x6	do.	Distiller & test tank.	2½	Main feed tanks.	2½	do.
						Reserve feed tanks.		
						Fresh water main.		
4	Forced lubrication.	6x7x8	do.	Lubricating oil cooler and drain tank.	3½	Main turbines.	3	In each ford engine room.
2	Oil cooler circulating	do.	do.	Sea.	3½	Oil cooler and drain tank.	3	In each ford engine room.
6	Fuel oil service.	5½x4x8	Vertical, piston, double acting, duplex.	Fuel oil main.	3	Burner supply	2½	In each boiler room.
4	Fuel oil booster.	6x8x8	Vertical piston, double acting, single.	Storage tank manifold.	3	Fuel oil main.	5	In Aft 1. boiler room, 2 in Aft 3. boiler room.
						Storage tank manifold.	5	
						Overboard. (5)	6	
2	Aux. condenser circulating.	3" Centrifugal. 26" impeller.	5x8 Engine	Sea.	8	Aux. condenser.	8	In each ford engine room.
2	Aux. condenser air.	6x10x8	Simplex, vertical, double-acting	Aux. condenser.	4½	Feed tank.	4	do.
4	Turbine drain.	3½x4x6	Horizontal, simplex.	Main turbine.	2	Filter tank.	1½	In under each turbine.
				Reserve feed tanks. (6)	1½	Wet air suction. (6)	1½	
3	Hand	2½x3½	Vertical, duplex.	Fuel oil suction pipe.	1¼	Burner supply.	¾	In each fire room.
3	Drainage	12" Centrifugal.	Submerged, direct connected, motor driven.	Main drain.	10½	Overboard.	12	In each fire room.
1	Drainage	8" Centrifugal.	Direct connected, motor driven.	Ford secondary drain.	4	Overboard.	3	Aux. pump room ford.
				Sea.	7	Flushing main.	5	
2	Fresh water.	3" Centrifugal.	do.	Fresh water tanks.	3	Fresh water main.	3	do.
2	Sanitary	3" Centrifugal.	do.	Sea.	5	Sanitary main.	5	All platform aft.

(1) Two suction pipes. (2) In No. 2. boiler room. (3) One pump in each after engine room. (4) In No. 1. boiler room only. (5) Port pump only. (6) Pumps in after engine room.

The main dimensions are as follows:

Diameter, inside shell, inches	48
Length overall, feet and inches	6–05
Coils, number	52
diameter, outside, inch	01
thickness, inch	00.065
Diameter feed inlet and outlet, inches	06
exhaust inlet, inches	08
drain, inches	02½

*Main Feed Pumps.**—Two Blake, vertical, double-acting, single, main feed pumps are installed on the center-line bulkhead of each forward engine room. Each pump has an independent suction from the feed-tank stand pipe cross-connection pipe and a discharge to the boilers through or by-passing the feed-water heaters.

*Main Fire and Bilge Pumps.**—Two Blake, vertical, double-acting, single, fire and bilge pumps are located on the center-line bulkhead of each after engine room. They are arranged to take suction from the sea or drainage system and discharge to the fire main, sanitary system or overboard. The forward pump in each engine room has a hose suction connection to the fuel-oil filling line and a discharge hose connection to the fuel-oil suction line.

*Turbine Drain Pumps.**—Under each main turbine there is a small Blake, horizontal, simplex pump with a suction from the turbine drains and a discharge to the filter tank and wet air cylinder of main air pumps. The pumps in the after engine rooms have a suction from the reserve-feed bottoms.

FORCED-LUBRICATION SYSTEM.

The forced-lubrication installation in each forward engine room consists of two forced-lubrication pumps*, one oil-cooler circulating pump*, one 250-gallon drain tank, one oil cooler, one 275-gallon oil-settling tank fitted with steam coils, and one 1,000-gallon oil-storage tank, with the necessary piping and fittings.

Either forced-lubrication pump draws oil from the drain tank and discharges into a 3-inch main through the cooler or by-passing same. From the 3-inch main a 2½-inch branch leads forward with branches to the high-pressure main and thrust bearings and the cruising-turbine bearings and gears. From the same main a 1¼-inch branch leads aft to the after engine room with branches to the low-pressure main and thrust bearings. Suitable channels are cut in the bearings

*See Table I.

to provide for a proper distribution of oil, the oil returning by gravity from the various bearings to the drain tank.

Either forced-lubrication pump can take oil from the drain tank and discharge to the settling tank or take oil from the settling or storage tanks and discharge into the system. The system may also be filled by gravity from the settling or storage tanks.

Cooling water for the oil cooler is supplied by the oil-cooler circulating-water pump or by a cross-connection from the water service.

Each main circulating pump is fitted with a self-enclosed forced-lubrication system and all spring bearings are fitted with a self-oiling system.

Data for One Oil Cooler.

Type	Reilly, vertical, multicoil.
Number of passes of oil	3
Diameter, inside, inches	40
Length overall, feet and inches	4–10
Coils, number	36
diameter, outside, inch	01
thickness, inch	00.065
cooling surface, square feet	221.4
Diameter, inlet and outlet, water, inches	03
oil, inches	03

WATER SERVICE.

A complete water-service system is fitted for supplying cooling water to all bearings if necessary. Water is supplied through a 3-inch pipe from the discharge of the main circulating pumps or through a 3-inch by-pass from the oil-cooler circulating pump. There is a 2-inch branch aft to the spring bearings and a 3-inch cross-connection to the oil cooler. Branches lead to all main and thrust bearings and a return from all bearings leads to the suction of the main circulating pump with a by-pass to bilge.

BOILERS.

There are twelve Yarrow water-tube boilers, arranged in three watertight compartments, as shown on Plate I. There are four boilers in each fireroom, arranged abreast.

The boilers are designed to operate the whole machinery installation at full power and are equipped for oil burning entirely.

The uptakes are the usual design and all combine into one smoke pipe at the second deck. The smoke pipe is 100 feet high above the base line and is rectangular at the second deck, changing to oval at the main deck. The smoke pipe is divided into three sections, one for the boilers in each fireroom, by means of division plates extending to the top of the stack. The net area of each section is 61.2 square feet.

Boiler Data.

Number	12
Pressure, working, pounds per square inch, gage	295
test, pounds per square inch, gage	450
Drum, steam, inside diameter, inches	50
length overall, feet and inches	12–07⅛
thickness, tube sheet, inches	01 15/16
wrapper sheet, inch	00½
water, number (elliptical) per boiler	2
length overall, feet and inches	11–00½
thickness, tube sheet, inches	01¾
wrapper sheet, inch	00 9/32
internal radius, circular part, inches	15½
Tubes, number per boiler	1,092
outside diameter, inches	01¾
thickness, mils	454 of 156½; 638 of 134
Height to center of steam drum from floor level, feet and inches	13–00
water drum from floor level, feet and inches	3–11
Distance center to center of water drums, feet and inches	11–10½
Heating surface per boiler, square feet	4,000
Furnace volume, cubic feet	625
Square feet of H.S. per cubic foot of furnace volume	6.40
Kind of forced draft	Closed fireroom.
Oil burners, number per boiler	7
type	Fore River.
Diameter of main steam stop valve, inches	06
main feed stop and check valves (2), inches	02½
auxiliary feed stop and check valve, inches	03
surface blow valve, inches	01½
bottom blow valve (2), inches	01½
safety valve, triplex, inches	04

The auxiliary feed enters the steam drum and the main feed enters the water drums, where it is baffled so that it rises through the two outer rows of tubes in each drum.

FUEL—OIL SYSTEM.

A complete oil-burning system of the Fore River mechanical atomization type is installed, with arrangements for air or steam atomization for use in port if required. The system consists of four booster pumps,* two in each No. 1 and No. 3 firerooms, starboard and port sides, respectively, six oil-service pumps,* two in each fireroom, six oil heaters, two in each fireroom, and the necessary strainers, piping, valves, etc. In addition, there is a hand pump* in each fireroom with a 1¼-inch suction line direct from oil bottoms C-92 and C-93 and a ¾-inch discharge to the burner main, for use when raising steam with no power on the ship.

The forward booster pumps draw oil from the forward oil bottoms and discharge to the booster mains. Similarly, the after pumps draw from the after oil bottoms and discharge to the booster mains.

Each forward booster pump has a 5-inch suction from the manifold on the same side of the ship as the pump. From each bottom 3-inch lines are run with a branch to each manifold and fitted with cutout valves, so that it is possible to take oil from any bottom from either manifold.

Each after booster pump has a 5-inch suction line running aft through the engine-room space, on the same side as the pump and cross-connected in the after engine rooms, with 3-inch branches to each oil bottom on that side.

All bottoms, except five, are fitted with two suction pipes, one of which leads to the lowest point of the bottom and the other to a point about six inches above the lowest point.

There are two 5-inch booster mains, one on each side, running fore-and-aft throughout the firerooms, into which the booster pumps discharge. The two mains are cross-connected in Nos. 1 and 3 firerooms.

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*See Table I.

From the booster mains 3-inch branches lead to the suctions of the oil-service pumps. Each service pump discharges into a 2½-inch main, and thence through or by-passing the oil heater, the two mains being cross-connected, and then divides into two 1¼-inch burner mains, one main running across the front of each boiler on that side. The burner mains are fitted with stop valves, operated from place and from the third deck.

There are seven burners per boiler, arranged in two rows, three in the top row and four in the bottom row. Three-eighth-inch lines lead from the burner main to the individual burners and are fitted with stop valves.

The whole system is completely equipped with stop and automatic valves as required.

The piping for the booster pumps is so arranged that oil may be drawn from any bottom and discharged to any other bottom on the opposite side of the ship for the after pumps, while the forward pumps can discharge to any other bottom that does not have a suction pipe in common with the one to the bottom from which oil is being taken. This applies to bottoms A-91, A-92 and A-93, which have a common suction pipe from the manifolds, with branches to the respective bottoms.

The fire and bilge pumps adjacent to the manifolds or filling lines have a portable connection for flooding the bottoms with salt water and the booster pumps have a portable connection for discharging overboard.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—There are twelve forced-draft blowers, four in each fireroom. The fans are of the multivane, double-inlet, single-width type built by the B. F. Sturtevant Co., and each has a rated capacity of 16,000 cubic feet of air per minute when discharging against a back pressure equal to five inches of water. They are located in special blower rooms, two blowers per room, on the first platform-deck level, and discharge directly into the firerooms through hinged

shutters that are arranged to close automatically in case the blower stops when the fireroom is under pressure.

Each fan is driven by a direct-connected, enclosed General Electric motor of 32 horsepower. The motors are controlled from the firerooms or the blower rooms at will. The blower data is as follows:

Revolutions per minute (variable)	725 to 930
Fan, diameter, inches	29½
width, inches	20

*Auxiliary Feed Pumps.**—There are three Blake, vertical, double-acting, single auxiliary feed pumps, one in each fireroom. They are arranged so that a pump can feed any boiler in the same compartment through the auxiliary feed line, or any boiler by by-passing to the main feed line.

*Fire and Bilge Pumps.**—There are three Blake, vertical, double-acting, single fire and bilge pumps, one in each fireroom. They are arranged to draw from the drainage system and sea and discharge to the fire main, sanitary system or overboard. The pump in No. 1 fireroom also has portable hose connections, suction from the oil-filling line and discharge to the fuel-oil manifold.

*Fuel-Oil Booster Pumps.**—There are two booster pumps in each Nos. 1 and 3 firerooms. They are Blake, vertical, double-acting, single type and are arranged to draw oil from the storage tanks and discharge to the booster mains, to the suction of the other booster pump in same compartment, and the pumps on the port side are arranged with a portable connection for discharge overboard.

*Fuel-Oil Service Pumps.**—There are two Blake, vertical, double-acting duplex service pumps in each fireroom, with suction from the booster mains and discharge to the burner lines.

Fuel-Oil Heaters.—There are two fuel-oil heaters in each fireroom for heating the fuel oil. They are of the following dimensions:

*See Table I.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	13
Length, overall, feet and inches	4–08½
Coils, number	3
diameter, outside, inch	01
thickness, inch	00.065
heating surface, square feet	18.45
Diameter, oil inlet and outlet, inches	02½
steam inlet, inches	01½
drain, inch	01

*Centrifugal Drainage Pumps.**—In each fireroom there is a 12-inch motor-driven centrifugal pump, with suction from the main drain and discharge overboard.

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel, as shown on Plate I. The lines are cross-connected in Nos. 1 and 3 firerooms and in the engine rooms by 8, 8½ and 9-inch connections, respectively. The branches from the boilers are 6 inches in diameter each, and the main lines are 8 inches in the forward fireroom, increasing to 10½ and 13 inches at each successive boiler connection.

AUXILIARY STEAM PIPING.

From the main steam cross-connection in No. 1 fireroom steam is led to the various auxiliaries in that compartment, to the forward dynamos, turbo air compressors, forward heating systems, and to the whistle and siren. In No. 2 fireroom there is a 3½-inch auxiliary steam cross-connection between the main steam lines, from which branches are led to the various auxiliaries in No. 2 fireroom. From the main steam cross-connection in No. 3 fireroom branches lead to the various auxiliaries in this fireroom. There are also two 8½-inch branches which lead aft through the after dynamo room and form a loop around the engine room from which branches are led to the after dynamos and auxiliaries, engine-room auxiliaries, after

* See Table I.

heating system, steam-steering engine, evaporators and ice machines.

Stop valves are fitted throughout where required.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe is fitted throughout the machinery spaces and elsewhere as required for the various auxiliaries. It is so arranged that the auxiliary exhaust may be directed into either the main or auxiliary condensers, either feed-water heater, into the atmosphere through two connections to the escape pipe in No. 2 fireroom, starboard and port sides, into the 5th stage of the high-pressure turbine or into the steam belt of the low-pressure turbine.

MAIN AND AUXILIARY FEED SYSTEMS.

Each main feed pump has a 5½-inch independent suction from the pipe connecting the standpipes on the two main feed tanks. The pumps discharge through the feed-water heaters, or by-pass same if desired, to the boilers. The combined discharge from each pair of pumps is 6 inches, uniting in No. 3 fireroom into a 7-inch connection leading forward and diminishing in size to 5 inches in No. 2 fireroom and to 4 inches in No. 1 fireroom. In each fireroom 4-inch connections lead to the two boilers on each side of the vessel with 2-inch branches to each water drum.

The auxiliary-feed suction main is 10 inches in diameter and takes off the stand pipe cross-connecting pipe. It runs forward through the firerooms, diminishing in size to 8 and 5½ inches, respectively, as the auxiliary feed-pump suctions take off. All pump suctions are 5½ inches in diameter. The discharge from the auxiliary feed pumps is 5 inches in diameter, with a 4-inch branch leading to the two boilers on each side. From the 4-inch lines 3-inch branches lead to the auxiliary feed-check valve on each boiler. The 5-inch discharge is cross-connected to the main feed line in each fireroom.

INTERIOR COMMUNICATION.

The usual engine-room and fireroom telegraphs, gong, telephones, voice tubes and fire-control instruments are fitted for transmitting orders and signalling to the machinery compartments and other parts of the vessel.

AIR–COMPRESSOR PLANT.

The compressors complete were built by the Ingersoll-Rand Co., and each is capable of supplying 2,350 cubic feet of air per minute at a pressure of 100 pounds per square inch when making about 6,000 revolutions per minute.

For running the pneumatic tools in the engineering department, blowing soot off the tubes, charging air chambers and atomizing oil, there are two 11-inch by 11-inch by 12-inch water-cooled Westinghouse steam-driven air compressors located in No. 2 fireroom. They are both connected to a reservoir of about 33,500 cubic inches capacity. The plant has a capacity of 90 cubic feet of air per minute at 150 pounds pressure. A pneumatic main runs from the reservoir throughout the machinery spaces with branches to the various tools and connections.

EVAPORATING AND DISTILLING APPARATUS.

This plant is located on the second platform deck, starboard side, outboard of the after dynamo room. It consists of four evaporators, two distillers, two feed-water heaters, two distiller circulating pumps, two evaporator feed pumps and two fresh-water pumps, together with the necessary piping and accessories. The plant has a rated capacity of 25,000 gallons of water per 24 hours and is arranged to operate in double effect.

Data for Each Evaporator.

Type	Reilly, horizontal, multicoil.
Diameter, inside, feet and inches	5–01½
Length, overall, feet and inches	5–02½
Coils, number	36
diameter, outside, inch	01
thickness, inch	00.065
heating surface, square feet	221.4
Diameter, steam inlet, inches (2)	03
vapor nozzle, inches	04½
feed valve, inches	01½
blow valve, inches	02½

Data for Each Distiller.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	31
Length, overall, feet and inches	4–03⅝
Coils, number	15
diameter, outside, inch	01
thickness, inch	00.065
Cooling surface, square feet	92.25
Diameter, circulating water inlet and outlet, inches	04½
vapor inlet, inches	04½
drain, inches	02

Data for Each Evaporator Feed-Water Heater.

Type	Reilly, U-type, Bureau standard.
Tubes, number of rows	4
per row	6
diameter, inside, inch	00⅝
thickness, inch	00.065
heating surface, square feet	23
Diameter, feed-water inlet and outlet, inches	02
vapor inlet and outlet, inches	04½

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward fireroom and one just abaft the after fireroom, located amidships on the second platform deck. The condensing apparatus is located in a separate room in the hold, directly underneath the dynamo rooms, and communicates with the latter by means of a ladder and access hatch.

There is one distribution room for each dynamo room, located directly overhead on the first platform deck. Each distribution room contains one combined switchboard which has panels for the lighting, power, searchlight and generator circuits.

There are two generators in each dynamo room, each driven by a 2-stage horizontal Curtis turbine. The generators are 6-pole, compound wound, 300-kilowatt, General Electric type. Each will deliver 2,400 ampères of current at 125 volts, when running at 1,500 revolutions per minute and is capable of delivering one-third overload for two hours without injury.

There is one condenser for each dynamo room, of the same general design as the main and auxiliary condensers. Each has an independent air pump, centrifugal circulating pump, hotwell pump and hotwell. The condensers can be used only in connection with the dynamos, except that the turbo-air compressors exhaust into the forward one.

Dynamo Condenser Data.

	Forward.	Aft.
Thickness of shell, inch	00 5/16	00 5/16
tube sheets, inch	01	01
Length between tube sheets, inches	88	88
Tubes, number	1,511	1,173
diameter, outside, inch	00⅝	00⅝
thickness, inch	00.065	00.065
cooling surface, square feet	1,813	1,407
Diameter, dynamo exhaust nozzles, (2) inches	20	20
room auxiliaries exhaust, inches	02½	02½
air compressor exhaust, inches	18	...
pump suction (combined), inches	06	06
circulating-water inlet and outlet, inches	10	09

Dynamo Condenser Air Pump Data.

Number	2
Type	Blake, vertical, twin-beam, single steam cylinder.
Diameter, suction nozzle (wet) inches	04
(dry) inches	04
discharge nozzle (wet), inches	04
(dry), inches	03½
steam cylinder, inches	08
water cylinders (2), inches	16
Stroke, inches	12

Dynamo Condenser Circulating Pumps and Engines, Data.

Type	Centrifugal, driven by a single engine.	
	Forward.	Aft.
Diameter, impeller, inches	26	26
suction, inches	10	09
discharge, inches	10	09
steam cylinder, inches	05	05
Stroke, inches	05	05
Revolutions per minute	385	380

GENERAL WORKSHOP.

The machine shop is located on the third deck, between the engine-room hatches, and is equipped with the following tools, each driven by an independent electric motor:

No.		Make and H.P. of motor.
1	28-inch by 51-inch swing, extension gap lathe, 10 feet between centers; The Putnam Machine Co.	Reliance, 7½ H.P.
1	14-inch Davis lathe, 24 inches between centers	Reliance, 1½
1	14-inch Davis lathe, 50 inches between centers	Reliance, 1½
1	Column shaper, 15-inch stroke by 15-inch traverse; Hendey-Norton	Reliance, 1
1	31-inch radial drill, capacity 1½-inch holes; The Dresses Machine Tool Co.	Reliance, 3
1	16-inch sensitive drill, capacity ½-inch holes; James Clark, Jr., Electric Co.	Reliance, ½
1	Universal milling machine, 22-inch longitudinal feed, 18-inch vertical movement, and 6-inch traverse; Hendey-Norton	Reliance, 2
1	Double emery grinder on column; Springfield Manufacturing Co.	Reliance, 3
1	Portable cylinder-boring machine, capacity 6 inches to 24 inches with 50-inch travel; Underwood	...
6	Reed vises, with jaws 5½ inches wide, opening 8 inches	...

BLACKSMITH AND COPPERSMITH SHOPS.

A blacksmith shop is provided on the third deck, port side, around the after uptakes; the coppersmith shop is in the same space on the starboard side. They are equipped with one portable and one permanent forge, an anvil, pipe-cutting

machine, and all necessary tools and fittings. The permanent forge is equipped with an electric-driven blast fan.

FOUNDRY.

The foundry is located on the third deck around the forward uptakes. It is equipped with a Bureau standard oil-burning crucible furnace and all necessary crucibles, tools, etc., for handling and pouring metal.

TORSION METERS, REVOLUTION INDICATORS, ETC.

In each forward section of line shafting there is installed a Gary-Cummings torsion meter. The shafting was calibrated with the meters in place and the following constants were obtained:

Constant for starboard shaft, 35.905172; port, 36.3557. S.H.P. = Constant × Meter Reading × R.P.M.

A complete Cummings Engine Log System is installed, including revolution indicators, averaging counters, stop clocks, course indicators, direction indicators, etc. The necessary instruments are installed in each engine room, chart house and conning platform.

REFRIGERATING PLANTS.

The refrigerating plant consists of two horizontal, steam-driven, Allen dense-air machines, manufactured by H. B. Roelker, of New York. Each machine is capable of producing the cooling effect of three tons of ice per day. The plant is located on the second platform deck, port side, between frames 76½ and 83, and is so arranged that either or both machines may be used on the refrigerating rooms, ice-making boxes or scuttle-butts.

There are four refrigerating rooms on the third deck, port side, frames 76 to 87. The crew's compartment serves as a vestibule for the butter and meat rooms. The officers' cooling room has no vestibule, its door opening direct to the passage way.

The ice-making boxes are located in the passage way outside the cooling compartments, and are of the standard type, capable of making eighteen cakes of ice at a time.

A hatch is provided in the meat room for charging from the main deck. The rooms are cork insulated in the usual manner.

VENTILATING SYSTEM.

Artificial ventilation is provided for all quarters, living spaces, passages, storerooms, magazines, machinery spaces, etc., as required. There are forty-eight ventilating fans and motors, exclusive of the forced-draft units. The fans were made by the B. F. Sturtevant Co., and the motors by the General Electric Co. Air is supplied on the plenum system. Toilet spaces and engine rooms are also supplied with exhaust fans.

Heater boxes are fitted in the ventilating ducts to all quarters below the upper deck, crew's spaces, etc., for heating the incoming air; no other means of heating these spaces is provided.

All motors are of the open type except those for turret ventilation and the motor on the exhaust fan located in the crew's washroom aft, which are of the enclosed type.

TRIALS.

Standardization Trial.—This trial was conducted on November 3, 1915, on the measured-mile course off Rockland, Maine. The weather conditions were very favorable. In all 23 runs were made over the course at various speeds, and from the data obtained it was found to require 211.75 revolutions per minute of the propellers to attain the designed speed of 20.5 knots; 194.35 r.p.m. for 19 knots; 150.15 r.p.m. for 15 knots, and 99.15 r.p.m. for 10 knots. The estimated displacement at the middle of the five high runs was 27,222 tons.

Table II contains the data obtained on the various runs, from which the curves, Plate VIII were plotted.

TABLE II - U. S. S. NEVADA
STANDARDIZATION TRIAL DATA - NOVEMBER 3, 1915.

No. of Run	Time on Course Mins.	Secs.	Speed in Knots	R.P.M. Star. Engine	Port Engine	Mean	S.H.P. Star. Engine	Port Engine	Total
1	5	41.5	10.54	103.77	106.07	104.92	1304	1427	2731
2	5	56.7	10.09	102.44	99.52	100.98	1324	1339	2663
3	5	52.5	10.21	99.42	99.62	99.52	1178	1231	2409
Mean of Group			10.23			101.59			2617
4	4	56.9	12.13	122.75	123.33	123.04	2116	2287	4403
5	4	44.8	12.64	122.69	124.41	123.55	2159	2307	4466
6	5	05.7	11.78	121.94	122.42	122.18	2102	2225	4327
Mean of Group			12.30			123.08			4416
7	3	45.5	15.96	153.29	155.18	154.24	4018	4513	8531
8	4	04.4	14.73	151.89	154.47	153.18	4036	4493	8529
9	3	47.2	15.85	153.39	153.75	153.57	4076	4360	8436
Mean of Group			15.32			153.54			8506
10	3	42.7	16.17	170.54	170.42	170.48	5940	6196	12136
11	3	25.7	17.50	171.14	171.29	171.22	5838	6165	12003
12	3	43.4	16.11	171.14	171.12	171.13	6022	6097	12119
Mean of Group			16.82			171.01			12065
13	3	03.8	19.59	194.98	194.36	194.67	9031	9186	18217
14	3	14.3	18.53	194.93	194.76	194.85	8889	9347	18236
15	3	02.9	19.68	196.08	196.62	196.35	9082	9579	18661
Mean of Group			19.08			195.18			18338
16	3	01.9	19.79	207.48	207.46	207.47	10951	11314	22265
17	2	55.4	20.52	207.69	208.69	208.19	10962	11532	22494
18	3	00.7	19.92	207.13	206.75	206.94	10932	11350	22282
Mean of Group			20.19			207.70			22384
19	2	52.0	20.93	215.86	214.98	215.42	12633	12896	25529
20	2	53.3	20.77	217.04	216.28	216.66	12858	13289	26147
21	2	53.0	20.81	215.84	214.98	215.41	12632	12818	25450
22	2	51.2	21.03	219.30	217.93	218.62	13465	13707	27172
23	2	51.1	21.04	220.12	218.61	219.35	13591	13670	27261
Mean of Group			20.90			217.02			26291

U.S.S. NEVADA
H.P.M. speed and power curves
H.P.M. 3 A.H.P.
H.P.M. 4 Motors
SCALE OF H.P.M.
100
110
120
130
140
150
160
170
180
190
200
210
220
230
SCALE OF SHAFT H.P.
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000
22000
24000
26000
SCALE OF KNOTS
9
10
11
12
13
14
15
16
17
18
19
20
21

TABLE III. TRIAL DATA U. S. S. NEVADA.

	12-Hour Full Speed Trial.	12-Hour Trial at 19-Knots.	12-Hour Trial at 15-Knots.	24-Hour Trial at 10-Knots.
Date of Trial	Nov.5, 1915	Nov.4, 1915	Nov.9, 1915	Nov.8-9, 1915
Speed in Knots	20.529	19.110	14.922	10.495
Draught, Mean, Feet and Inches	28 - 2 3/16	28 - 2 5/8	28 - 2 7/8	28 - 2 3/4
Displacement (Corresponding), Tons	27,137	27,177	27,204	27,194
Number of Boilers Used	12	12	8	4
Burners (7 per Boiler)	81.51	57.92	24.00	11.71
Nozzles, Each H.P.Turbine	19	16	---	---
Cruis. "	---	---	14.75	6.00
Heating Surface Used, Square Feet	48,000	48,000	32,000	16,000
per S.H.P. "	1.992	2.605	4.105	5.683
Cooling Surface, Sq.Ft. per S.H.P. "	.822	1.074	2.540	7.034
PRESSURES, LBS. PER SQ.IN.(AVERAGE)	Star. Port	Star. Port	Star. Port	Star. Port
Main Steam at Boilers Gauge	284.22	285.88	283.93	276.36
Engine-Rooms "	258.21	265.55	272.63	266.73
Cruis.Turb. Steam Chest Abs.	---	---	266.00 | 269.50	271.21 | 273.79
1st Stage "	---	---	68.08 | 69.67	26.02 | 28.35
H.P.Turb. Steam Chest "	264.31 | 264.40	238.38 | 239.54	--- | ---	--- | ---
1st Stage "	77.71 | 79.83	62.00 | 62.54	22.96 | 23.17	11.06 | 11.29
Drum Head, Forward "	42.38 | 45.40	32.88 | 35.42	12.17 | 15.06	5.54 | 6.10
Aft "	28.17 | 31.83	22.46 | 24.50	9.75 | 9.83	5.10 | 4.44
L.P.Turb. Drum Head "	14.90 | 15.03	13.04 | 11.96	5.38 | 5.00	3.25 | 2.83
Exhaust Shell "	2.00 | 2.00	1.67 | 1.50	1.83 | 1.52	1.38 | 1.27
Vacuum, Main Cond., Inches of Hg.	27.50 | 27.50	27.93 | 27.92	28.48 | 28.38	28.29 | 28.25
Barometer, Inches of Hg.	30.07	30.15	29.94	30.15
Auxiliary Steam Gauge	273.61	280.96	284.67	277.61
Exhaust "	4.99	4.96	4.40	4.78
Gland Steam "	4.82	3.64	4.60	3.90
Main Feed Discharge "	338.40	308.00	334.58	323.75
Forced Lubrication "	58.34	56.71	61.55	61.06
Fuel Oil to Burners "	198.60	200.00	204.00	229.30
Air in Firerooms, Inches of Water	4.20	3.59	2.31	2.32
TEMPERATURES, DEGREES, FAHR.(AVERAGE).				
Outside Air	49.08	49.08	53.17	47.79
Engine-Rooms, Forward	90.27	88.54	92.21	84.65
Aft	71.90	69.50	76.17	72.86
Firerooms	98.58	95.39	98.80	97.00
Main Injection	51.42	51.00	52.00	51.42
Outboard Delivery	75.42 | 72.63	76.42 | 74.58	69.92 | 70.67	68.63 | 71.21
Main Air Pump Discharge	62.00 | ---	62.92 | ---	63.46 | ---	62.17 | ---
Feed Water, Measuring Tanks	77.38 | 80.17	71.67 | 81.50	78.33 | 80.92	87.92 | 81.63
from Heater	97.38 | 141.13	112.92 | 153.58	147.50 | 184.00	--- | 172.83
Oil to Cooler	100.38 | 97.08	98.25 | 94.17	100.58 | 110.50	89.83 | 95.83
from Cooler	81.21 | 77.71	79.83 | 75.83	87.00 | 92.67	79.38 | 84.00
Uptake	468.00	396.00	370.00	310.00
Oil Drains,Cruis.Turbs., Forward	--- ---	--- ---	152.33 | 126.00	115.54 | 114.04
Aft	--- ---	--- ---	121.70 | 155.00	108.83 | 119.13
Gear Thrust	--- ---	--- ---	94.58	83.88
H.P.Turbines, Forward	94.90	91.42	102.00	92.90

From Cooler		81.21	77.71	79.83	75.83	87.00	92.67	79.38	84.00
Uptake		468.00		396.00		370.00		310.00	
Oil Drains, Cruis. Turbs.,	Forward	---	---	---	---	152.33	126.00	115.54	114.04
	Aft	---	---	---	---	121.70	155.00	108.83	119.13
	Gear Thrust	---	---	---	---	94.58		83.88	
H.P. Turbines,	Forward	94.90		91.42		102.00		92.90	
	Aft	95.10		92.44		99.00		90.90	
L.P. Turbines,	Forward	105.65		101.25		106.80		92.90	
	Aft	100.60		97.60		100.40		90.90	
Fuel Oil to Heaters		69.74		66.48		62.54		64.08	
Burners		132.70		131.60		124.40		122.86	
R.P.M. of Forced Draft Blowers		916.80		809.50		794.00		788.65	
DOUBLE STROKES PER MINUTE OF PUMPS									
Main Circulating		196.58	220.38	144.83	161.08	116.00	138.33	100.17	117.46
Air		24.79	25.63	21.83	25.17	23.58	23.33	21.79	19.79
Feed		15.37	14.63	24.83	19.58	10.21	9.75	---	9.92
Forced Lubrication		27.21		26.13		46.80		40.96	
Oil Cooler Circulating		31.40		23.75		43.63		26.29	
Fuel Oil Booster		48.96		41.92		31.00		Intermittent	
Service		17.97		12.43		8.80		8.97	
R.P.M. Starboard Shaft		211.66		195.56		149.92		105.10	
Port Shaft		212.53		195.49		148.82		103.63	
Mean		212.10		195.53		149.37		104.37	
Slip of Propeller, Starboard		18.28		17.67		16.14		15.87	
Port		18.44		17.46		15.34		14.49	
Mean		18.36		17.57		15.74		15.18	
Shaft Horse Power, Starboard Shaft		11,164		8,704		3,654		1,227	
Port Shaft		12,148		8,933		3,718		1,258	
Total		23,312		17,637		7,372		2,485	
from Curve		24,100		18,430		7,795		2,815	
WATER CONSUMPTION, ALL MACHINERY									
Lbs. per Hour, Evaporated		371,145.42		297,749.33		127,867.25		64,595.08	
per Sq. Ft. H.S.		7.732		6.203		3.996		4.037	
Lb. of Oil		11.647		12.45		13.069		13.35	
per S.H.P.*		15.400		16.156		16.404		22.947	
Knot Run		1,808		1,558		856.8		615.5	

NOTE.—*S.H.P. used is that from Standardization Curve.

STEAM TURBINE BLADE FASTENINGS.

By Jas. A. Capstaff, New York Shipbuilding Company.

The results of a test on the Vickers-Ferranti blade fastening were given on page 329 of the May issue of the Journal, 1915. Later it occurred to Mr. Lovekin that it would be interesting to have an absolutely true comparison of the strengths of the Vickers-Ferranti, Brown-Boveri and Lovekin methods carried out under the same conditions.

Accordingly, test pieces of the Lovekin and Brown-Boveri fastenings were made up and pulled to destruction.

Fig. 1 shows the test piece of Vickers-Ferranti fastening as tested and dealt with in the article of last May.

Fig. 2 shows the test piece of the Lovekin fastening, and Fig. 3 shows the test piece of the Brown-Boveri fastening.

From these it will be seen that in all cases the blades were of mild steel and of the same dimensions and the caulking strips were of brass. The same distance pieces and holder stops were used. The holders were similar, the only difference being the method of securing the blades.

In the Vickers-Ferranti test a load of 10,700 pounds was required to pull the blade out of the holder. Examination showed that the metal composing the dovetail of the blade was drawn almost parallel and the metal forced into the vee serrations in caulking was sheared off. The caulking piece remained in the holder.

In the Lovekin test a load of 17,300 pounds was required to pull the blade out of the holder. In this test the metal at the root of the recess of the blade was sheared off. The locking strip remained in the holder and was locally indented, in way of the blade, about .03 inch.

In the Brown-Boveri test a load of 11,100 pounds was required to pull the blade out of the holder. The metal at the root of the recess of the blade was sheared off, and the metal forced into the vee serrations in caulking was sheared off.

The results of the above tests show the Lovekin blade fastening to be 61½ per cent. stronger than the Ferranti method, Fig. 1, and 56 per cent. stronger than the Brown-Boveri method, Fig. 3.

In my previous article I mentioned the fact that the Lovekin fastening was particularly suitable for impulse blades.

Figs. 4 to 10 show the Lovekin method of fastening applied to Parsons standard reaction blades. The depth of the grooves are the same as used by Messrs. Parsons.

Fig. 11 is a photographic reproduction of a model of the fastening assembled, and Fig. 12 shows very clearly the shapes of the various pieces.

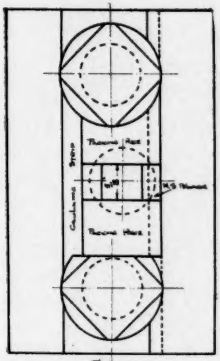
In regard to reaction-blade fastenings it is to be remembered that the blades are of a finer section than impulse blades, and therefore do not lend themselves very well to the majority of blade fastenings which could be used with fair success on impulse blades.

The well known Parsons method of securing the blades by caulking the distanct pieces between is, indeed, a good fastening for reaction blades when carried out by experienced and reliable bladers.

A 250 B blade held in the Parsons standard fastening and caulked by an experienced blader requires a load of about 1,250 pounds to pull it out, which is truly a good figure.

The writer made up a test piece of the Lovekin method applied to a 250 B standard Parsons blade, and it withstood a load of 1,900 pounds, which showed it to be 52 per cent. stronger than Parsons.

It is a well known fact that brass expands more than steel or iron; therefore any method of blading, where brass is used, that has the distance pieces caulked tightly into grooves or projections in the adjoining blades, must form a continuous



PLAN.

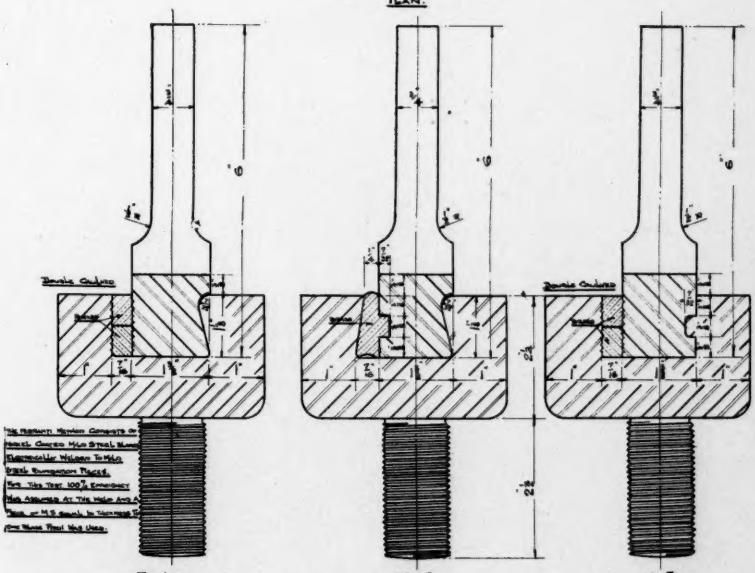


FIG. 1.
TEST Nº 13.
VICKERS – FERRANTI.
MAXIMUM PULL 10,700 LBS.

FIG. 2.
TEST Nº 20.
LOVEKIN
17,300 LBS.

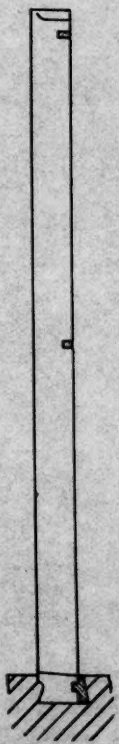
FIG. 3.
TEST Nº 21.
BROWN – BOVERI.
11,100 LBS.

COMPARATIVE TESTS OF BLADE FASTENINGS.

SCALE 3"=1'0"

280 s.
280 d.
No 10.

270 s.
270 d.
No. 9.



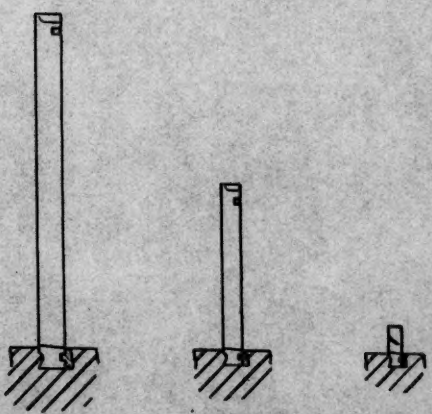
260 s.
260 d.
No. 8.

Rupee Gaspoits Kept Sunk Depot as T

250 s.
250 d.
No 7.

240
240
No 6

[illegible] Depth as Records Specimens.



240 b. 130 b. 120 b. Elevation.
240 c. 130 c. 120 c. Section.
Fig 6. Fig 5. Fig 4.

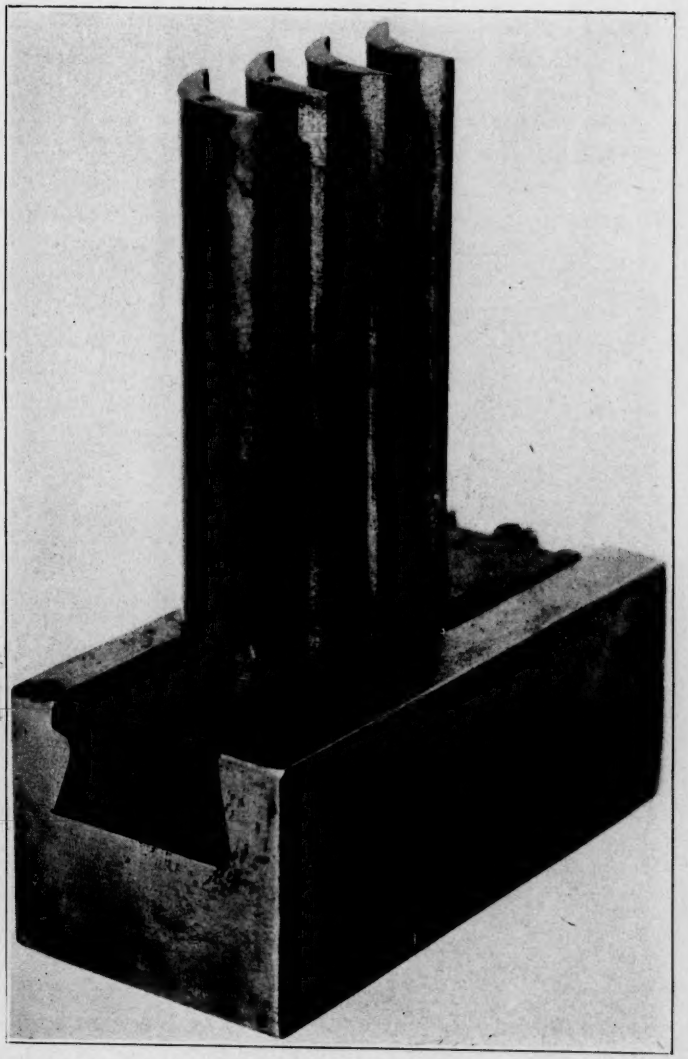


FIG. 11.

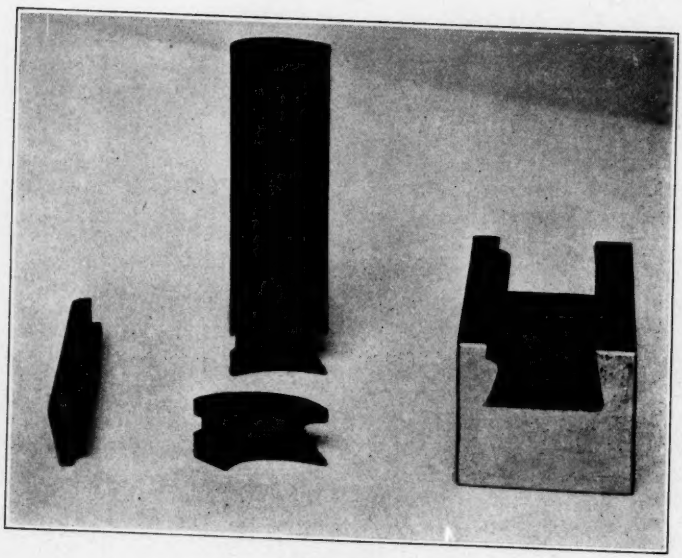


FIG. 12.

ring of brass within the groove on the steel rotor or the iron casing. Now, if the groove is completely filled and heat applied, the brass ring " composed of blading and distance pieces" must expand, and, inasmuch as there is no room left between the blades and distance pieces for expansion, the compression of the metal must necessarily force the blading outward and thereby reduce the clearance between the blade tips and the rotor or casing. This is likely to cause blade stripping or, at the least, rubbing. The Lovekin method positively avoids this trouble, on account of the space left between the blades and distance pieces being ample for the increased expansion, where brass blades and distance pieces are used.

The Lovekin method further permits of a uniform spacing and absolute radial alignment of blades around the entire circumference, and avoids large gaps or openings between the final blades. The reason for this is that all blades are inserted loosely around the entire circumference until the proper number of blades are inserted, then the radial gage is applied so as to insure radial alignment of the blades, and, finally, the side locking ring is caulked solidly, which holds all blades in their proper places.

Brass locking rings can always be used regardless of the material used in the blades and distance pieces, because we have room at the ends of lacing bars for expansion.

STOPPING, BACKING AND TURNING SHIPS.

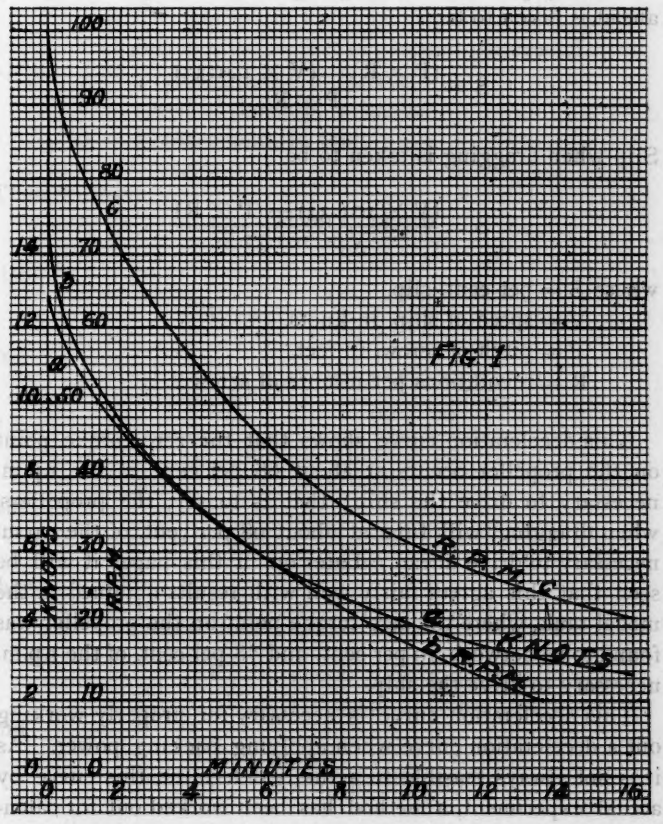
By Lieut. S. M. Robinson, U. S. N., Member.

The data given in this paper was principally obtained in connection with the development of electric drive for ships. The performance of an induction motor is vitally affected by the performance of the ship so that in addition to the normal "steaming-ahead" condition there are three others that must be considered. These are (1) stopping (that is motors running free with no power on them), (2) backing (with ship going ahead), (3) turning. In the past, little attention seems to have been paid to these points, so it was necessary to do considerable experimenting in order to determine what actually happens in each of these cases.

STOPPING.

In the case of a ship fitted with reciprocating engines, when the signal "stop" is received the engines are held stopped; if it is necessary to do so, the links are thrown over and enough steam admitted on the backing side to hold the engines stopped. In this case the screws act as a powerful brake, and stop the ship rapidly. In the case of turbine ships, some of them shut steam off the ahead turbines and let the propellers keep revolving ahead while others admit steam to the backing turbines to hold the screws stopped; however, it is not believed that the latter practice is much used. An electrically-propelled ship is similar to the turbine ship when the latter uses no steam in the backing turbine. The deceleration of a reciprocating-engine ship will therefore be considerably more rapid than either a turbine ship or an electrically-propelled ship when the engines are stopped.

For the purpose of determining what this deceleration is, experiments were conducted on the *Jupiter* by running over a measured course with power off and propellers running freely. Observations were taken on shore and also on board



ship, and from these were plotted speed and r.p.m. deceleration curves. These are shown in Fig. 1. Curve *a* represents the actual speed of the ship at any time interval, curve *b* represents actual revolutions per minute of the screws at any

time interval, and curve *c* represents the revolutions per minute necessary to drive the ship at the speeds represented by curve *a*. From curve *b* it will be seen that the slip of the screws, when dragging, is about 28 per cent. and from curve *c* that it is about 9 per cent. when going ahead. Slip when going ahead is taken to be,

$$S = \frac{(P \times R) - (V \times 101.33)}{P \times R}.$$

Slip when dragging is taken to be

$$S = \frac{(V \times 101.33) - (P \times R)}{V \times 101.33},$$

where S = per cent. slip.
V = speed of ship, in knots.
P = pitch of screw, in feet.
R = revolutions per minute of screw.

This deceleration curve shows that the zero torque point on the propeller occurs at about 68 per cent. of the r.p.m. necessary to drive the ship; that is to say, the propellers, when dragging, will turn at about 68 per cent. of the r.p.m. necessary to drive ahead. Later on in this paper it will be seen that this agrees fairly well with results obtained in the model tank, where the zero torque point on the propeller was found to be between 70 per cent. and 78 per cent. of the r.p.m. necessary to drive the ship.

This deceleration curve is necessary for properly working out a deceleration curve when applying backing power. As it is not always feasible to actually determine this curve by actual experiment, a method has been worked out for calculating it, and it is believed that it will be accurate enough for all practical purposes.

By this method the deceleration curve can be obtained whether the ship is running with engines stopped or backing. The following is the method used:

Let H.P. = horsepower exerted at any instant to stop the ship.
W = work done (per second) by this horsepower.
F = force in pounds acting on the ship to stop it.
M = mass of the ship.
a = deceleration in knots per hour per minute.
a' = deceleration in feet per second per second.
g = force of gravity = 32.16.
V = speed of the ship, in knots.
D = displacement, in tons.

Then we have

$$W = \frac{\text{H.P.} \times 33{,}000}{60},$$

also

$$W = F \times \frac{V \times 6{,}080}{60 \times 60},$$

or

$$\frac{\text{H.P.} \times 33{,}000}{60} = \frac{F \times V \times 6{,}080}{60 \times 60};$$

$$\therefore F = \frac{\text{H.P.} \times 33{,}000 \times 60}{V \times 6{,}080}.$$

But

$$F = Ma' = \frac{D \times 2{,}240}{32.16} \times a',$$

then

$$\frac{\text{H.P.} \times 33{,}000 \times 60}{V \times 6{,}080} = \frac{D \times 2{,}240 \times a'}{32.16},$$

$$\therefore a' = \frac{\text{H.P.} \times 33{,}000 \times 60 \times 32.16}{V \times D \times 2{,}240 \times 6{,}080},$$

and

$$a = \frac{\text{H.P.} \times 33{,}000 \times 60^4 \times 32.16}{V \times D \times 2{,}240 \times 6{,}080^2}.$$

For the *Jupiter* D was 16,670 tons at the time of the experiment, so for that ship we have

$$a = \frac{\text{H.P.} \times .009964}{V}.$$

To calculate the H.P. acting to stop the ship at any speed V, there will have to be added together the effective horsepower necessary to drive the ship at the given speed and the horsepower due to the braking effect of the screw if the ship is running with power shut off or the power delivered by the engines if she is backing. If the ship is running without power it is believed that the following method of estimating the braking effect of the screw will be accurate enough. Consider the action of the screw (while revolving freely) to be similar to that of the struts. This seems a reasonable assumption, as the screws will at all times have water back of them if they are revolving. As an example of this method take the *Jupiter*.

Strut area (one), ... = 8.96 sq. ft.
Propeller hub area (one), ... = 7.92 sq. ft.
Total area covered by strut and propeller hub = 16.88 sq. ft.
Projected area of propeller (one), ... = 60.56 sq. ft.

From Captain Dyson's propeller charts we find the strut resistance to be 9.3 per cent. of the resistance of the bare hull.

$$\therefore \quad \text{Propeller resistance} = \frac{60.56}{16.88} \times 9.3 = 33.7 \text{ per cent.}$$

of bare hull resistance.

From Captain Dyson's charts we find total appendage resistance to be 11.3 per cent. of bare hull resistance.

Total added resistance will be

11.3 per cent. + 33.7 per cent. = 45 per cent.

$\therefore$ Total H.P. = 1.45 × effective horsepower (bare hull).

In Table 1 the values of a have been calculated by substituting these values of H.P. in the equation previously derived. The derived values of a are shown in Fig. 2. In this figure it will be seen that the curve between any two speeds differing by only one knot is nearly a straight line, so that the average deceleration, while the ship is dropping one knot, can

TABLE I.—"JUPITER," STOPPING.

Knots.	E.H.P. (Bare Hull).	H.P. = 1.45 E.H.P.	a	Time to decelerate 1 knot.	Total time from 21 knots.
21	.....	.....	*14.0	*.0	*.0
20	.....	.....	*11.0	*.0800	*.08
19	11,800	17,100	9.0	.1000	.18
18	8,950	12,970	7.21	.1235	.3035
17	6,525	9,450	5.56	.1565	.4601
16	5,150	7,470	4.67	.1860	.6461
15	4,075	5,910	3.94	.2320	.8781
14	3,200	4,640	3.31	.2755	1.1536
13	2,500	3,630	2.79	.3280	1.4816
12	1,950	2,830	2.36	.3878	1.8694
11	1,500	2,175	1.98	.4560	2.3254
10	1,125	1,630	1.63	.556	2.8814
9	825	1,195	1.33	.676	3.5574
8	575	834	1.04	.840	4.3974
7	425	616	.88	1.042	5.4394
6	255	370	.616	1.338	6.7774
5	170	246.5	.493	1.8	8.5774
4	98	142	.355	2.36	10.9374
3	50	72.4	.242	3.34	14.2774
2	25	36.3	.181	4.94	19.2174
1	10	14.5	.145	6.14	25.3574
0	0	0	.0	∞	∞

*Obtained by extending the curve.

be taken as the average of the deceleration at the two speeds. Using this method, the time for the ship to drop to any speed has been calculated in Table 1. This gives a deceleration curve which has been plotted on Fig. 3. The actual deceleration curve (obtained by experiment) is also shown in this figure. This curve can be represented very closely by an equation of the form—

$y(x+b)=a$. In this case the equation is of the form $y=\frac{52.92}{x+2.52}$, where y = speed of the ship and x = time intervals. It will be seen that the calculated, actual and equation curves all agree very closely. It will be noted that at

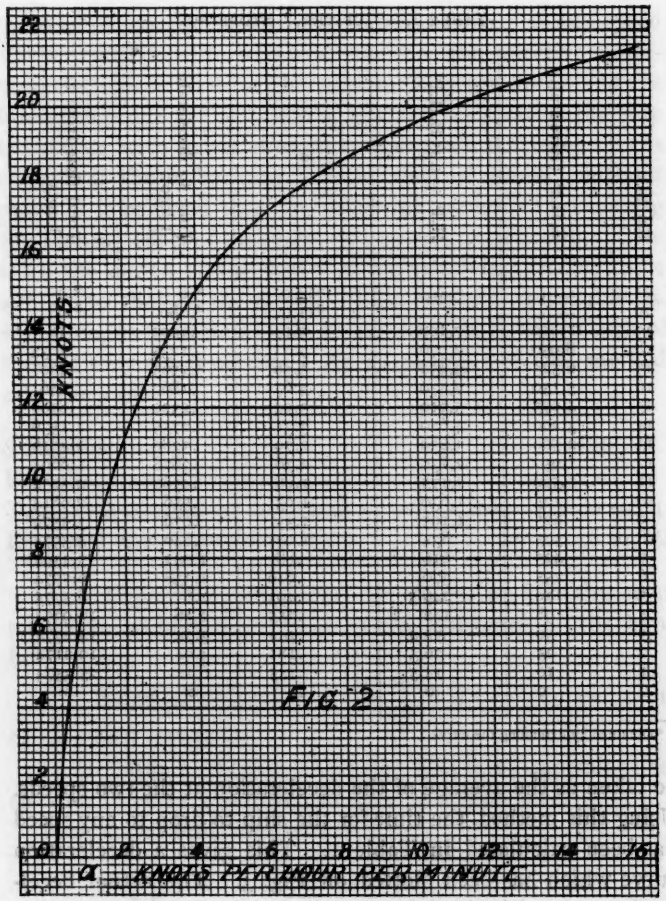


FIG 2

the high speeds the change of speed is much more rapid than at the low speeds.

Following the above method, the deceleration curves for the *California* have been determined, and these will be used later on in the paper when the subject of "backing" is treated.

California's strut area (two on one side of the ship) = 13.3 square feet.

Area of two propeller hubs = 11.94 square feet.
Total strut area = 13.3 + 11.94 = 25.24 square feet.
Projected area of two propellers = 107.7 square feet.
Strut resistance = 9.4 per cent.

Propeller resistance $= 9.4 \times \frac{107.7}{25.24} = 40.2$ per cent.

Appendage resistance = 14.9 per cent.
Total added resistance (to bare hull) = 14.9 + 40.2 = 55.1 per cent.

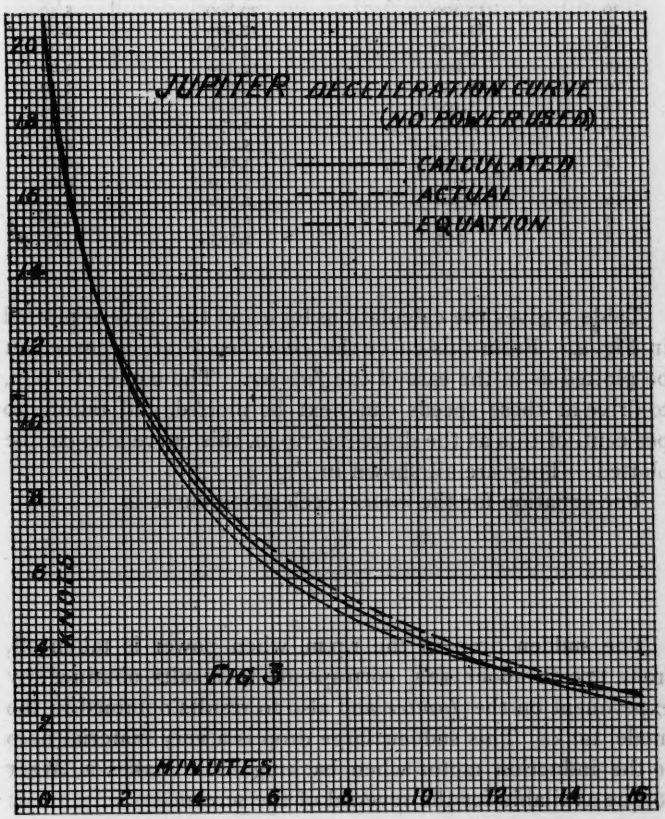


FIG 3

Substituting D = 32,000 tons in the a equation previously reduced, we have

$$a = \frac{\text{H.P.} \times .005191}{\text{V}}.$$

TABLE 2.—("CALIFORNIA," STOPPING).

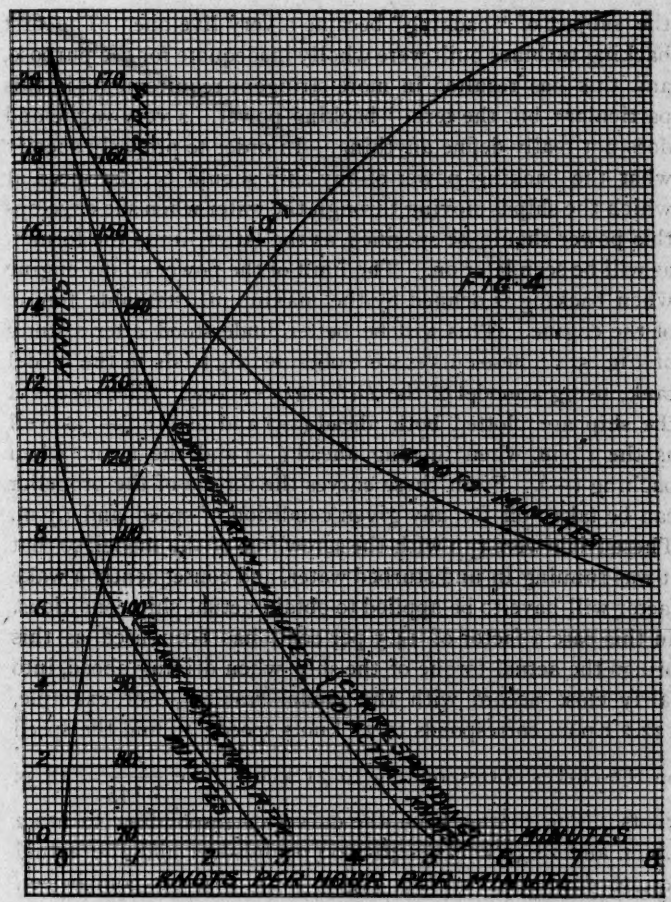
Knots.	E.H.P. (Bare Hull).	H.P. = E.H.P. × 1.551.	a
22	21,200	32,900	7.77
21	16,300	25,300	6.25
20	13,400	20,800	5.40
19	11,200	17,400	4.75
18	9,400	14,600	4.21
17	7,800	12,100	3.69
16	6,350	9,850	3.19
15	5,100	7,920	2.74
14	4,100	6,370	2.36
13	3,250	5,050	2.02
12	2,550	3,960	1.71
11	1,950	3,030	1.43
10	1,500	2,330	1.21
9	1,100	1,710	.986
8	775	1,205	.781

Table 2 gives the calculations for a and the curve is plotted in Fig. 4. From this curve the *knots* deceleration curve can be plotted as in the case of the *Jupiter*. The curve of r.p.m. to drive at these speeds can next be plotted, and taking 70 per cent. of this as the dragging r.p.m. this curve also can be plotted. They are all shown in Fig. 4.

The equation for the *knots* deceleration curve is

$$y = \frac{81.4}{x + 3.87}.$$

The sudden drop in the r.p.m. when power is taken off, from 175 to about 122, corresponds to results obtained by experiment on the *Jupiter* and also to results obtained in the model tank. This sudden large drop is a very material help to the induction motor when backing as it makes a larger torque available for the reversal of the screw.



BACKING.

The subject of "backing" seems to have been very little considered in the past. The main reason for this probably was that with reciprocating engines the backing power was ample and was fixed by the design of the engine itself. When the marine turbine first entered the field of marine propulsion

the subject of "backing" became a very live one. The first backing turbines built were wholly inadequate for the purpose, and this has resulted in more stringent requirements in this particular; but the term "backing power" is very vague and does not really define anything. In order to properly specify what the backing power of any ship should be, the speed at which the ship is moving through the water when she develops this power should be specified, as the latter limits the former, as will be seen later on. The limit of the power of any engine when backing is defined by the maximum attainable torque of the engine. This will be better understood after a study of Fig. 5. These curves were obtained by trials in the model tank, using a model of the *Delaware's* screw. The speeds of the ship are plotted both ahead (+) and astern (—) as well as the r.p.m. when turning ahead (+) and the r.p.m. when backing (—). From these curves the torque of the propeller can be taken off for any given condition of speed and r.p.m. These trials were run with the screw free of the model, so that it was running in undisturbed water, and consequently a wake factor will have to be applied to obtain actual ship conditions. In this case a factor of 14.5 per cent. has been used, as this brings the actual torque of the ship, when driving ahead, into fairly close accord with the model results. For example, 122.2 r.p.m. corresponds to 21 knots speed; to find the speed on the curves corresponding to 21 knots, take $\frac{21}{1.145} = 18.35$ knots. The torque, from the curves corresponding to 122.2 r.p.m. and 18.35 knots is 465,000 pounds-feet and the actual torque developed by the engine was 464,500 pounds-feet. Using this same method for all speeds Table 3 has been prepared, and this shows fairly close accord between actual engine torque and propeller torque. However, it is not intended to use this model-tank curve for actual values but only for comparative ones.

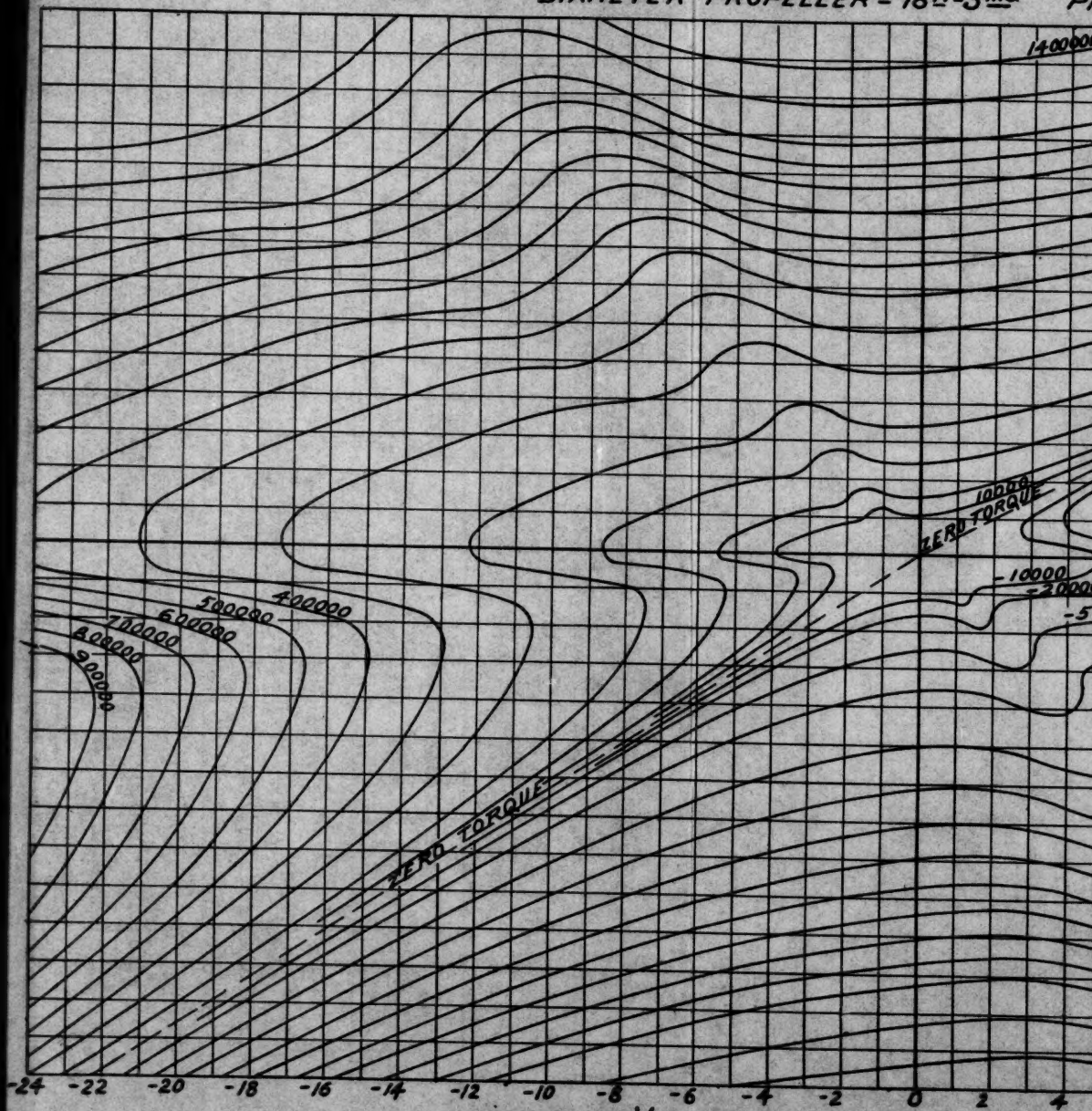
There are two very striking phenomena to be noticed about these curves. The first is that, with speed of ship constant, the torque of the propeller, as its revolutions per minute are

CONTOUR CURVES OF TORQUE OF THE U.S.S. DE

EXPRESSED IN POUND FEET FOR ONE PROPELLER DERIVED FROM EXPE

BASED ON ASSUMPTION THAT LAW OF COMPARISON HOLDS FULLY AN

DIAMETER PROPELLER = 18 FT-3 INS P



SCALE FOR V_A, SPEED OF ADVANCE IN KNOTS

U.S.S. DELAWARE PROPELLER

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INS. PITCH = 19FT - 9INS.

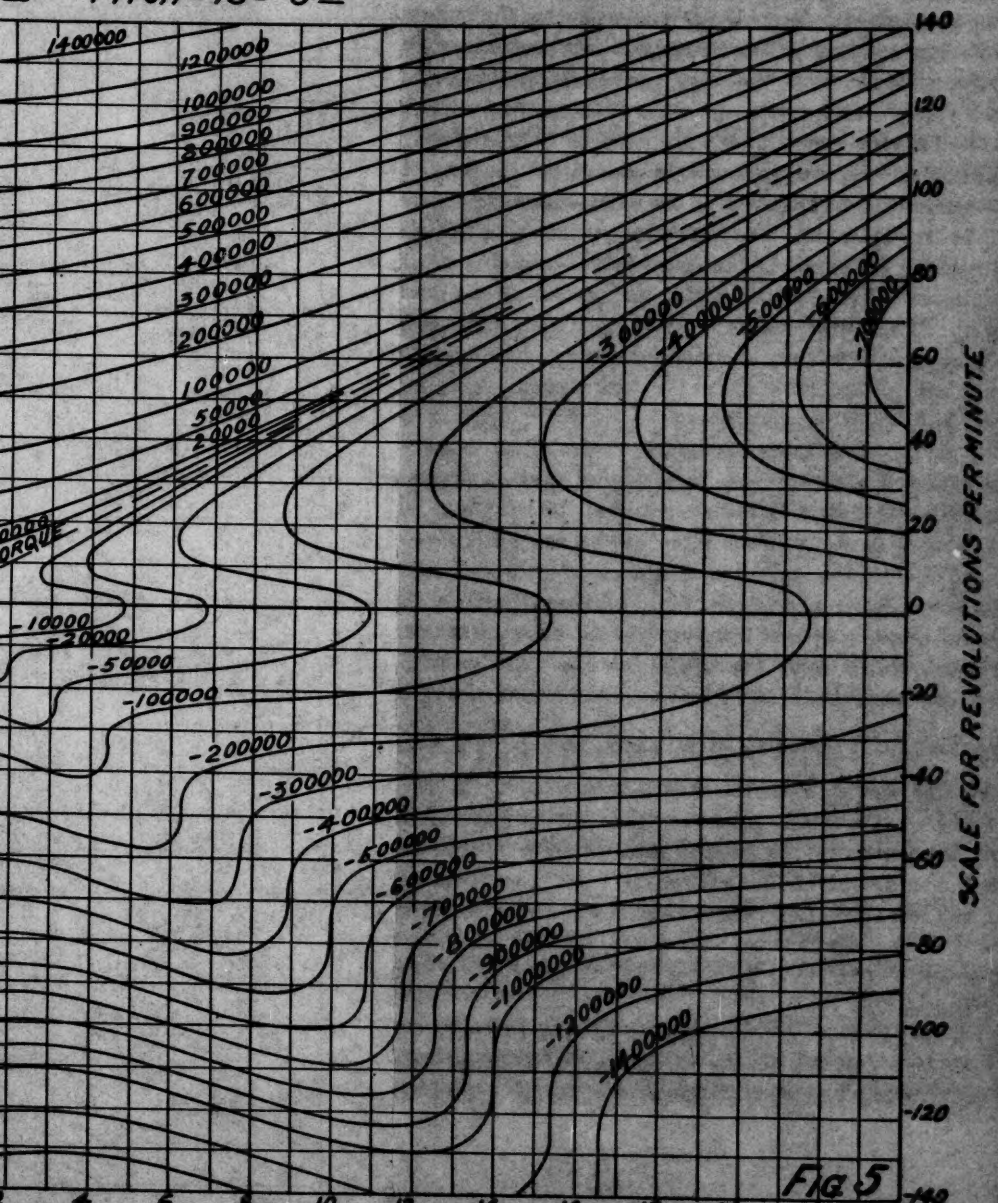
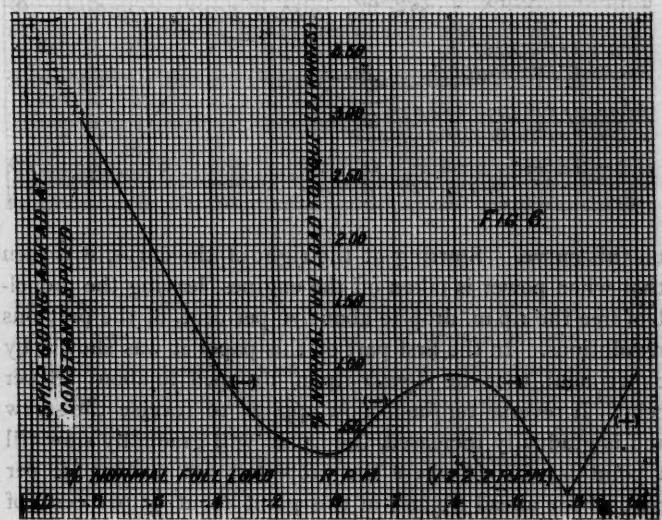


FIG 5

TABLE 3.—"DELAWARE."

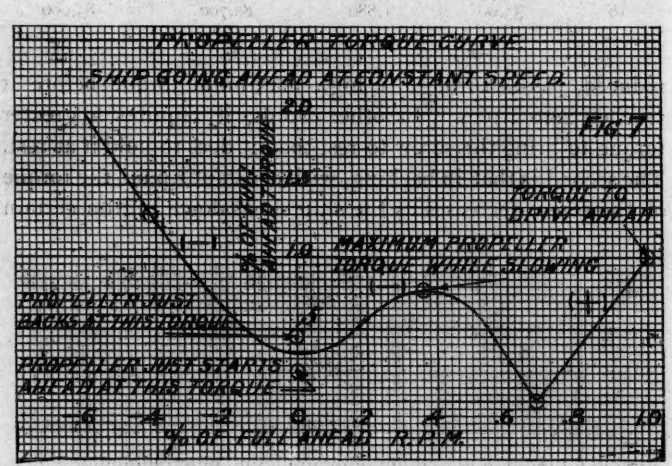
Knots.	R.p.m.	I.H.P.	Actual torque.	Curve torque (14.5 factor).
21	122.2	23,450	464,500	465,000
20	113.75	18,100	385,000	380,000
19	107.3	14,700	331,500	332,000
18	101.5	12,200	290,500	292,000
17	95.75	10,250	258,800	255,000
16	90.0	8,600	231,000	230,000
15	84.25	7,050	202,300	201,500
14	78.25	5,700	176,300	175,500
13	72.50	4,550	151,800	145,500
12	66.60	3,550	129,000	125,000
11	60.75	2,650	105,400	100,000
10	55.00	1,880	82,700	83,000

reduced, passes through a high maximum torque before it reaches zero r.p.m. In other words, it requires a greater torque to bring the screw to rest than it does to hold it at rest. The second is that in backing, with constant r.p.m., the torque of the screw decreases as the ship slows down until a certain



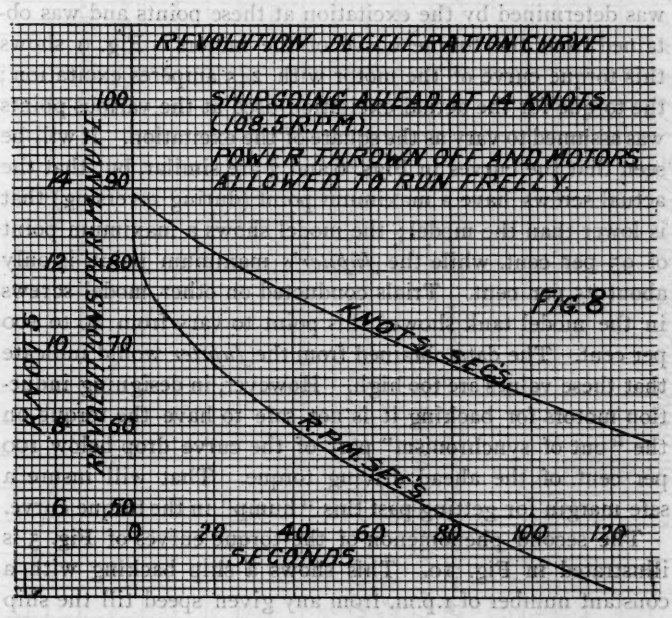
speed is reached, when the torque begins to increase; it reaches a maximum and then decreases again before the ship becomes stopped. Both of these phenomena have been verified by actual experiments on the *Jupiter*. They will each be taken up and considered in detail.

To illustrate the first point Fig. 6 has been plotted from the curves in Fig. 5. The ordinates of this curve represent per cent. of the normal ahead driving torque and the abscissae represent per cent. of ahead r.p.m. corresponding to the speed which it is assumed the ship is making. The ship is assumed to be making a constant speed ahead at all points represented


PROPELLER TORQUE CURVE
SHIP GOING AHEAD AT CONSTANT SPEED.
FIG 7
TORQUE TO DRIVE AHEAD
MAXIMUM PROPELLER TORQUE WHILE SLOWING
PROPELLER JUST BACKS AT THIS TORQUE
PROPELLER JUST STARTS AHEAD AT THIS TORQUE
% OF FULL AHEAD R.P.M.

on this curve. Starting at the right of the curve, it is seen that when power is taken off the engines, leaving the propellers free the r.p.m. drop to about 76 per cent. of the previous revolutions. In the first part of this paper it was shown, by experiment, that the *Jupiter* r.p.m. dropped to about 68 per cent. If reverse torque is now applied to reverse the screw and is gradually increased, the r.p.m. will gradually slow till a point is reached where the propellers are making 40 per cent. of the ahead r.p.m.; at this point about 95 per cent. of the ahead torque will be required; from this point on down

to standstill less torque will be required to slow the propeller; when the propeller becomes stopped the torque has reached a minimum, and will rise again if the screw is actually reversed. The curve given was plotted for the condition of ship going ahead at 21 knots, but it is approximately correct for all speeds, as will be seen by following out the various speeds in Fig. 5. In Fig. 7 is given a similar torque curve for the *Jupiter*. This curve was determined by experiment in the following manner. With the ship going ahead at 14 knots power was suddenly



thrown off; the propeller speed dropped to the point marked zero torque. The excitation of the generator was then reduced as much as possible and the backing switches thrown in; the propellers kept revolving ahead and the excitation was gradually increased till the propellers just passed over the maximum torque point and started to reverse; the excitation was then reduced to just enough to keep the propeller stopped. There were two points for the stopped condition,

one at which the propeller would just start revolving ahead and the other at which the propeller would just back; the curve has been run between the two points. At each point the elapsed time from the beginning of the experiment was taken and, by means of the deceleration curves in Fig. 8, all points were reduced to the same speed. The curves in Fig. 8 were made up from "dragging" data taken at the time and from the deceleration curve given in Fig. 1. The torque at the two points, that is the maximum and minimum points, was determined by the excitation at these points and was obtained from the torque curve of the motors. Fig. 9 shows this torque curve of the motor with 245 ampères excitation; the torque for the actual excitation used at the various points was assumed to vary as the square of the excitation. It will be seen that the curves in Figs. 6 and 7 are similar, but that the actual screws have a maximum point (during reversing) that is lower than the model; the model shows a maximum point of 95 per cent. while the *Jupiter's* maximum point is only about 75 per cent. Trials conducted on other model screws in the model tank showed this point to vary from 80 to 100 per cent. The data obtained from the *Jupiter* would indicate that these values are too high. However, in designing induction motors for backing it is not safe to have the torque on the "out of synchronism" part of the curve drop below 100 per cent. of the ahead driving torque. That will insure a safe margin for getting past this "hump" in the torque curve.

The second phenomenon of the torque curves of Fig. 5 is illustrated in Fig. 10. This shows a ship backing with a constant number of r.p.m. from any given speed till the ship is stopped. The r.p.m. assumed are those which will give 100 per cent. ahead driving torque at the instant of backing. This curve is approximately correct for all speeds. From it we see that the torque necessary to turn the screw at the given r.p.m. falls as the ship slows till about 22 per cent. speed is reached, when the torque begins to rise and continues to rise till 5 per cent. speed is reached, when it begins to fall again. This curve was also verified by experiment on the

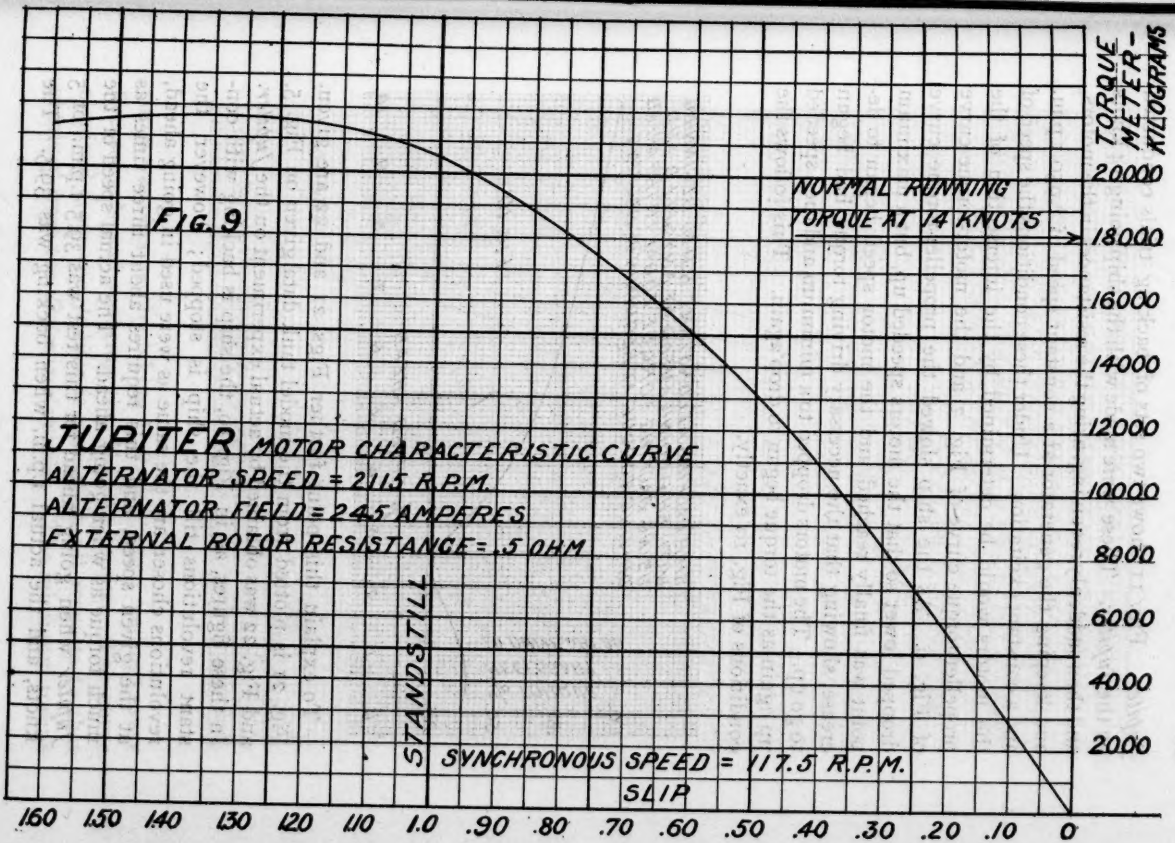


FIG. 9
JUPITER MOTOR CHARACTERISTIC CURVE
ALTERNATOR SPEED = 2115 R.P.M.
ALTERNATOR FIELD = 24.5 AMPERES
EXTERNAL ROTOR RESISTANCE = .5 OHM
NORMAL RUNNING TORQUE AT 14 KNOTS
TORQUE METER-KILOGRAMS
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
STANDSTILL
SYNCHRONOUS SPEED = 117.5 R.P.M.
SLIP
1.60 1.50 1.40 1.30 1.20 1.10 1.0 .90 .80 .70 .60 .50 .40 .30 .20 .10 0

Jupiter. Fig. 11 shows two sets of backing trials conducted on the *Jupiter*. These were made with the ship going 14 knots and then suddenly reversing, using the resistances in the motors, and keeping the generator at a constant speed of 1,950 r.p.m. and a constant excitation. Under these conditions the speed of the motors would be determined by the intersection of the propeller-torque curve of Fig. 7 and the motor-torque curve of Fig. 9. As the ship slowed the propeller-torque curve dropped lower so that the motors speeded up, but a maximum point was finally reached and the motor speed began to decrease, showing that the necessary driving torque had begun to go up. The motors dropped to a minimum and then speeded up again as the torque began to drop again. This follows the conditions of Fig. 10 exactly.

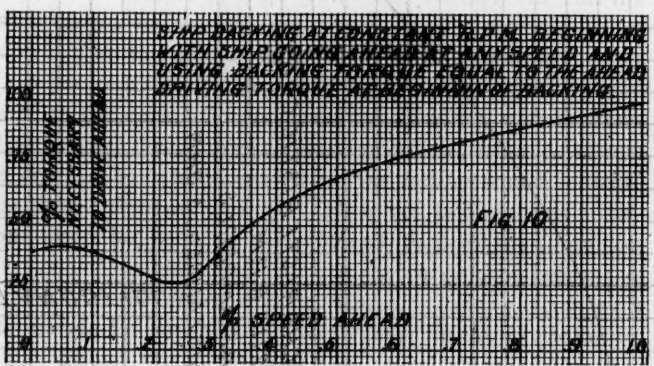


Fig. 10.

To explain this point further Figs. 21 and 22 are given. Fig. 21 is plotted from the model tank data given on Fig. 5, and Fig. 22 was obtained by actual experiment on the *Jupiter*. In these figures, as in Fig. 10, the ship is backing with constant revolutions till the ship is stopped; however, the revolutions chosen are the same as were used in going ahead, at the given speed, and this requires about three times as much torque as when going ahead. The actual speed of the *Jupiter* when going ahead for this test was 39.5 r.p.m., or 5 knots, and the actual r.p.m. when backing was 39.5. The

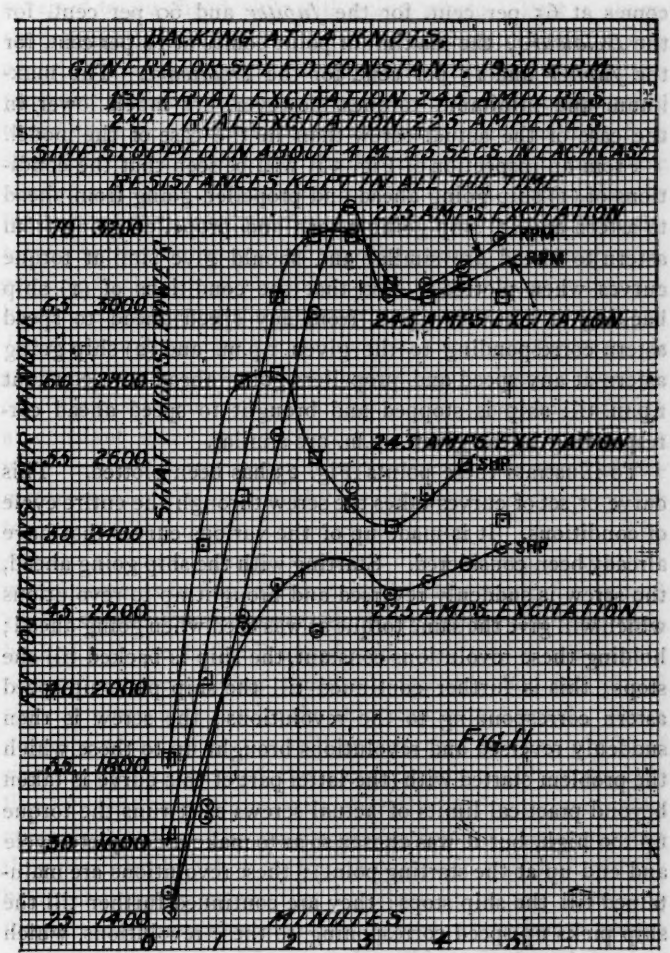


Fig. 11.

curve was obtained by taking H.P. and r.p.m. readings every five seconds and reducing the H.P. to torque. The curve obtained from the *Jupiter* is similar in all respects to that obtained from the *Delaware's* model. The minimum torque

comes at 65 per cent. for the *Jupiter* and 60 per cent. for the *Delaware;* the maximum point comes at 12 per cent. for the *Jupiter* and 15 per cent. for the *Delaware.* The maximum and minimum torques, however, are much lower in the case of the actual screw than in the case of the model.

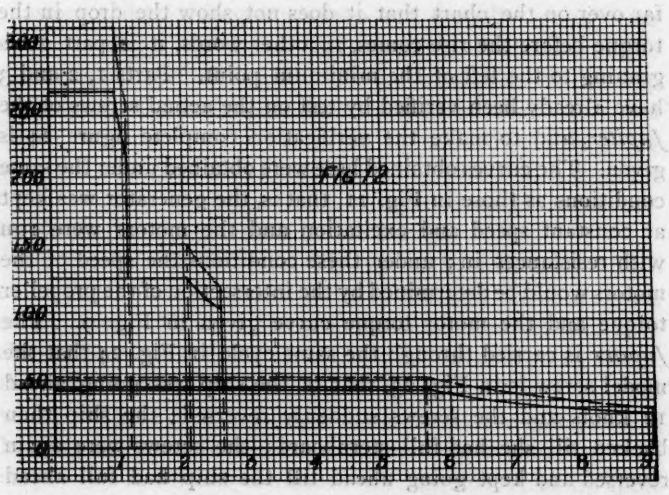
From an inspection of Fig. 5 it will be seen that the conditions of (1) ship going ahead and propeller going from ahead to astern and (2) ship going astern and propeller going from astern to ahead are similar and should give similar torque curves when plotted. Also that the conditions of (1) ship backing at constant r.p.m. from any speed ahead to speed astern corresponding to the given r.p.m. and (2) ship going astern at any speed and propellers going ahead at a constant r.p.m. till ship is stopped and brought to speed ahead corresponding to the given r.p.m. are similar.

To illustrate these points Fig. 23 has been plotted. This curve or set of curves takes the screw through the entire cycle of conditions. It is made up of the various curves that have already been considered. Starting with the ship going ahead, the screw is suddenly reversed and brought up to revolutions which will give the same torque as was used when going ahead; holding these revolutions constant, the ship is backed till she stops; this is further continued till the ship picks up speed astern corresponding to the revolutions; the screw is then suddenly reversed and revolutions brought up to those which the problem started with (the latter part of this curve is taken beyond practical limits of actual screws as it runs the torque up too high, but it was chosen so as to make a complete cycle and end up at the starting point); these revolutions are maintained till the ship stops; they are continued further till the ship picks up speed corresponing to these revolutions, which brings us back to the starting point. Part 1 shows the cycle through which the torque passes while the screw is being reversed; part 2 shows the change in torque while the ship is slowing down, backing at constant r.p.m. Part 3 shows the ship picking up sternboard with screw going at same r.p.m. Part 4 shows the torque cycle of the screw when it is suddenly

reversed to go ahead, ship still going astern. Part 5 shows the torque cycle while the ship is slowing down, propeller going ahead at constant r.p.m. Part 6 shows the torque cycle of the propeller while ship is picking up speed ahead, propeller going ahead at same r.p.m. as before. From inspection it will be seen that parts 1 and 4 are similar curves, parts 2 and 5 are similar and parts 3 and 6 are similar. Part 5 was taken so far over on the chart that it does not show the drop in the torque before the rise comes, in other words, it is part 2 beginning to the left of the minimum point. Parts 1, 2 and 3 have already been verified by test on the actual screws of the *Jupiter*, and to make the verification complete figure 24 is given. The curves obtained here were obtained under the same conditions as those in Fig. 11, that is, the generator was kept at constant speed and excitation and the motors were run with resistances in; under these conditions the speed of the motors would be determined by the intersection of the propeller torque and the motor torque curve given in Fig. 9. The *Jupiter* is carried through the same cycle in Fig. 24 that the model screw is in Fig. 23, that is, the ship was going ahead 12 knots and the screws suddenly reversed; the ship then backed till she had full sternboard; the screws were again reversed and kept going ahead till the ship had full ahead speed. The points where the ship stopped are noted. From an inspection of Fig. 9 it will be seen that on the right-hand side the torque curve of the motor is practicably a straight line, so that revolutions vary directly as the torque. The curves plotted in Fig. 24 are revolutions of the screw, but they may also be taken as torque on the motor shaft simply by reversing the curves, that is, when r.p.m. are increasing torque is decreasing, when r.p.m. reaches a maximum torque reaches a minimum, and so on. It will be seen that the curves are similar to parts 2, 3, 5 and 6 of Fig. 23; also the two parts of Fig. 24 are similar. This confirms the correctness of the shape of all the curves given in Fig. 23.

Now that it has been shown how a propeller acts during the entire cycle of backing, from the instant the power is

removed till the ship is stopped, some cases of actual backing will be taken up. As previously stated, the power any engine is capable of delivering while backing is limited by the maximum torque of the engine. To make this plainer the engines of the *Delaware* are taken as an example, and two theoretical indicator cards have been constructed and are shown in Fig. 12. The heavy-line curve shows the card



when going at full power, the data for the card being taken from the full-power trial.. The dotted card shows the conditions if full boiler pressure could be obtained in the high-pressure valve chest. In the construction of these cards no account has been taken of wire-drawing or clearances, as they are only for the purpose of illustration. The data used in the construction of the cards is as follows:

Diameter H.P. cyl.	= 38.5 inches.
Diameter I.P. cyl.	= 57.0 inches.
Diameter L.P. cyl. (2)	= 76.0 inches.
Stroke	= 48 inches.
H.P. cut off	= .86 inch.
I.P. cut off	= .8 inch.
L.P. cut off	= .62 inch.

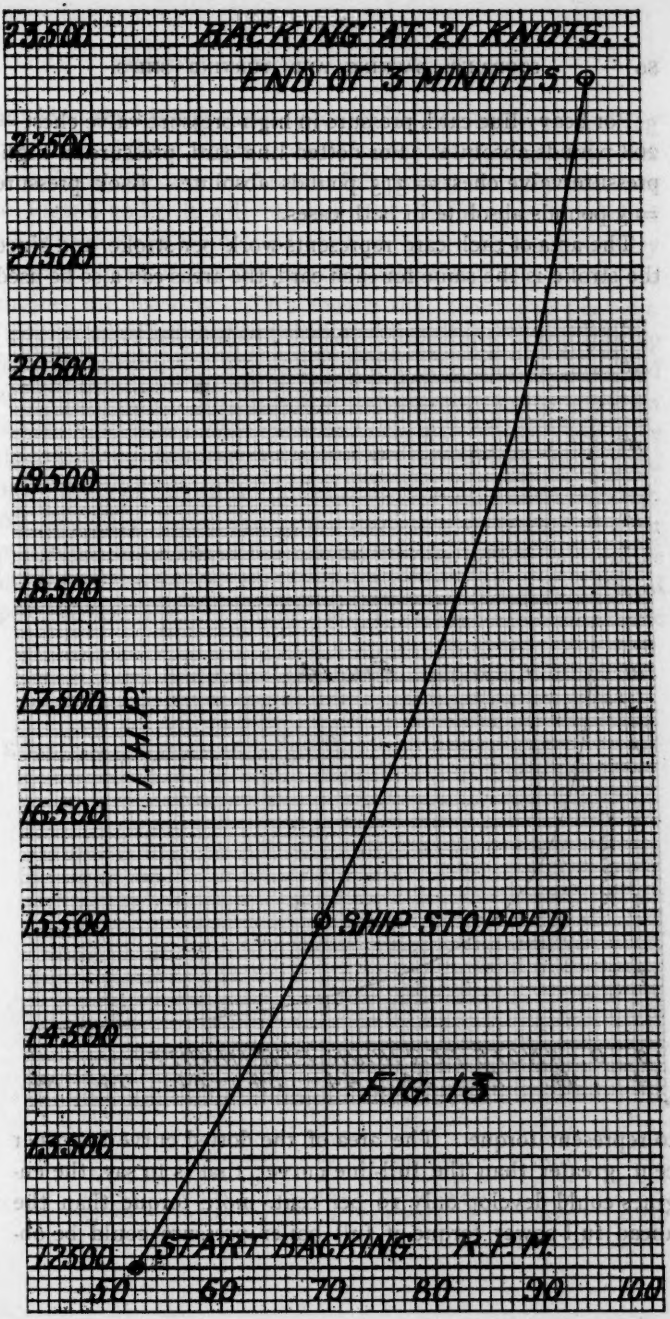
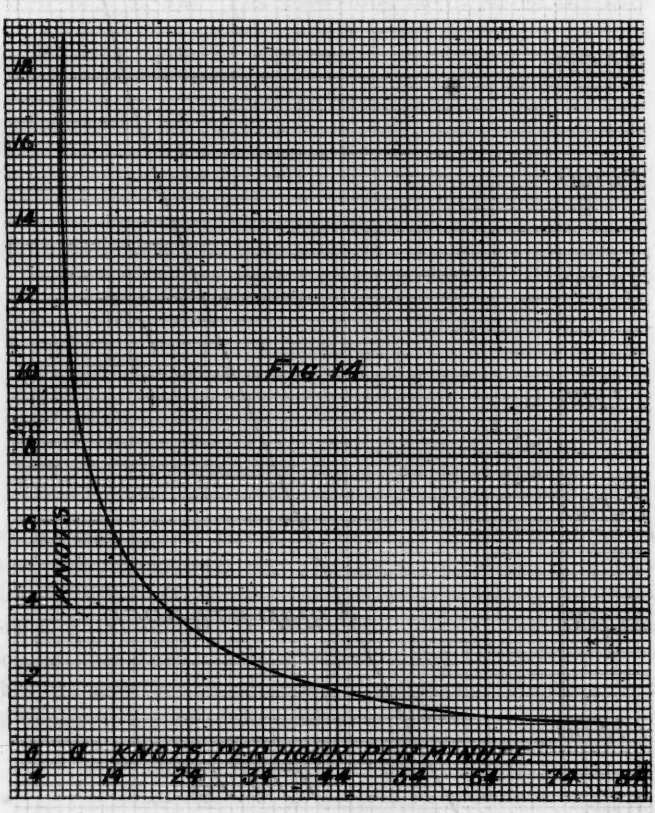

23500
BACKING AT 21 KNOTS.
END OF 3 MINUTES
22500
21500
20500
19500
18500
17500
16500
15500
SHIP STOPPED
14500
13500
12500
START BACKING
R.P.M.
50
60
70
80
90
100

FIG. 13

For heavy-line card, pressure in high-pressure valve chest = 268 pounds absolute. For dotted line card, pressure in high-pressure valve chest = 315 pounds absolute. Back pressure = 5 pounds absolute in both cases.

The area of each card represents work per stroke, and since the stroke is the same for each card, the areas can also be used



to represent torque. The area of the dotted curve is 19 per cent. greater than the full-line curve; that is to say, the engines could develop only 19 per cent. more torque than the ahead full-power torque if full boiler pressure could be ob-

tained in the high-pressure valve chest. Actually, on backing trials, the highest torque attained was about 9 per cent. greater than the torque developed on the full-power trial. By reference to Figs. 6 and 7 it will be seen that this torque is reached when backing at about 40 per cent. of the revolutions necessary to drive ahead, if the ship is going ahead full speed and the engines are backing. This means that about 43.6 per cent. (1.09 × 40) of full ahead power will be developed when the ship first begins to back. This amount will, of course, be increased as the ship slows. In the case of turbine ships the torque is far less than in the case of reciprocating engines, probably not more than half, so that they probably do not develop more than one-quarter of full ahead power at beginning of their backing. An induction motor can be designed to give a much greater maximum torque than the normal driving torque; also, since induction motors for ship propulsion will have two sets of pole connections, the motors can be arranged to back on the

TABLE 4.—"DELAWARE," BACKING.

Knots.	I.H.P.	I.H.P.×P.C.	E.H.P. (all app.)	Total H.P.	a	Interval time in minutes.	Total time.
21	.....	.....	.....	.....	.....	.....	.....
19	12,500	8,170	9,900	18,070	7.9	.2925	.2925
18	12,665	8,275	8,500	16,775	7.75	.1276	.4201
17	12,825	8,400	7,225	15,625	7.65	.1299	.5500
16	12,990	8,500	6,000	14,500	7.53	.1315	.6815
15	13,150	8,600	4,900	13,500	7.48	.1331	.8146
14	13,315	8,720	3,900	12,620	7.49	.1335	.9481
13	13,475	8,820	3,100	11,920	7.62	.1322	1.0803
12	13,640	8,920	2,430	11,350	7.85	.1292	1.2095
11	13,800	9,030	1,880	10,910	8.24	.1242	1.3337
10	13,965	9,130	1,400	10,530	8.75	.1176	1.4513
9	14,125	9,250	1,050	10,300	9.50	.1095	1.5608
8	14,290	9,350	750	10,100	10.50	.1000	1.6608
7	14,450	9,450	500	9,950	11.82	.0897	1.7505
6	14,615	9,560	300	9,860	13.65	.0784	1.9289
5	14,775	9,660	190	9,850	16.35	.0667	1.8956
4	14,940	9,770	120	9,890	20.55	.0542	1.9498
3	15,100	9,880	60	9,940	27.50	.0416	1.9914
2	15,265	9,980	20	10,000	41.50	.0290	2.0204
1	15,430	10,100	10	10,110	84.00	.0160	2.0364
0	15,600	10,200	0	10,200	∞	.0076	2.0440

slow-speed connection. This will allow the turbine to run at nearly full speed while backing at a low number of propeller r.p.m. This condition is ideal for getting high power while backing; the motor is capable of producing large torque and the turbine is running at a sufficiently high speed to enable it to develop full power. In other words, the turbine condition when backing on the slow-speed connection is practically the same as when going ahead at full power on the high-speed connection. This should give very fine backing results.

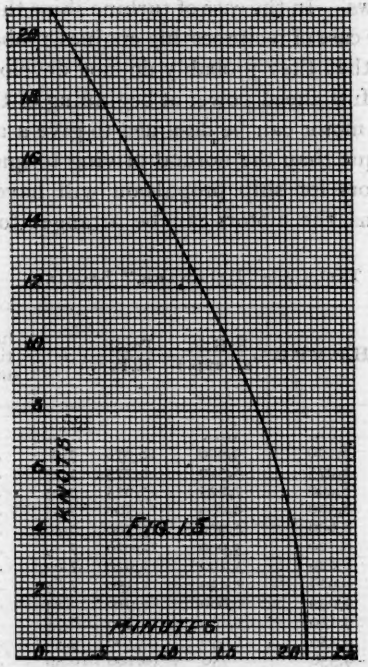


FIG. 13

To show what this means a comparison has been made with the *Delaware* when she was going ahead at 21 knots and the engines were suddenly reversed. Fig. 13 gives the data obtained on this trial. At the beginning she developed 12,500 I.H.P., which is 43.8 per cent. of her ahead full power, and at stop she developed 15,600 I.H.P., which is 54.6 per cent.

of her full power. The *Delaware's* displacement is 20,000 tons. Substituting this value of D in the a equation, we have for the *Delaware*.

$$a = \frac{\text{H.P.} \times .00831}{\text{V}}.$$

Substituting the data given by Fig. 13 in this equation, Table 4 has been calculated for the backing condition. The values of a obtained have been plotted on Fig. 14.

From the values of a given on this curve a deceleration curve has been plotted on Fig. 15. This curve shows the total time to stop the ship to be 2 minutes 3 seconds. The actual time as measured was 2 minutes 21 seconds. The results are considered to be very close, as it is difficult to determine the exact instant a ship becomes dead in the water. The same method of calculation has been followed for the *California*. In her case D = 32,000, so we have

TABLE 5.—"CALIFORNIA," BACKING.

Knots.	E.H.P. (B.H.)	E.H.P. (all) 14.9 per ct.	S.H.P. × P.C.	Total H.P.	a	Mins. (interval.)	Time in Mins.
21	16,300	18,740	0,000	.....	.....	0	0
20	13,400	15,400	18,740	34,140	8.87	.232	.232
19	11,200	12,880	18,740	31,620	8.65	.114	.346
18	9,400	10,800	18,740	29,540	8.53	.1165	.4625
17	7,800	8,960	18,740	27,700	8.45	.1178	.5803
16	6,350	7,300	18,740	26,040	8.45	.1182	.6985
15	5,100	5,860	18,740	24,600	8.52	.1178	.8163
14	4,100	4,710	18,740	23,450	8.70	.1162	.9325
13	3,250	3,730	18,740	22,470	8.97	.1132	1.0457
12	2,550	2,930	18,740	21,670	9.38	.109	1.1547
11	1,950	2,240	18,740	20,980	9.90	.1036	1.2583
10	1,500	1,720	18,740	20,460	10.62	.0975	1.3558
9	1,100	1,260	18,740	20,000	11.55	.0843	1.4401
8	775	890	18,740	19,630	12.75	.0823	1.5224
7	525	600	18,740	19,340	14.35	.0738	1.5962
6	325	375	18,740	19,115	16.55	.0647	1.6609
5	175	200	18,740	18,940	19.70	.0551	1.7160
4	75	86	18,740	18,826	24.40	.0453	1.7613
3	50	58	18,740	18,798	32.55	.0351	1.7964
2	25	29	18,740	18,769	48.75	.0246	1.8210
1	10	12	18,740	18,752	97.50	.0137	1.8347
0	0	0	18,740	18,740	∞	.00530	1.8400

$$a = \frac{\text{H.P.} \times .00591}{V}.$$

Table 5 shows the results of the calculations for a. It has been assumed that full power is developed all the way through due to the large maximum torque of the induction motor. It

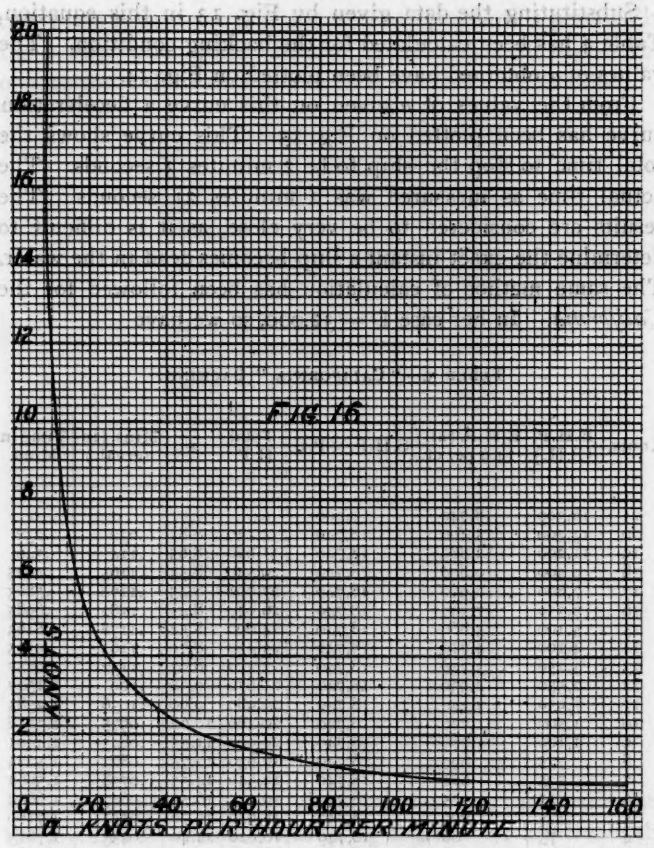


FIG. 16.

has also been assumed that the ship drops in speed to 20 knots during the act of reversal. The values of a have been plotted on Fig. 16 and from these values the deceleration curve has

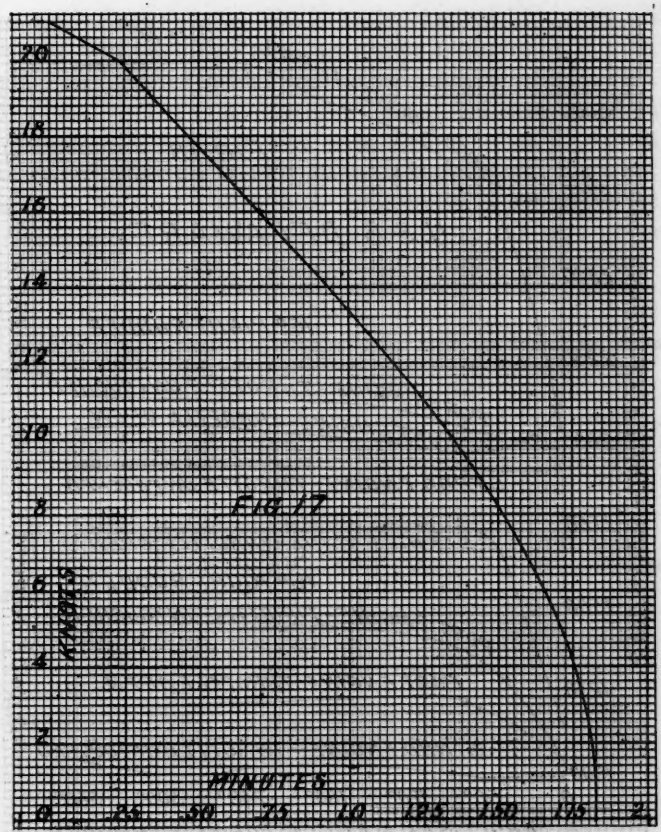
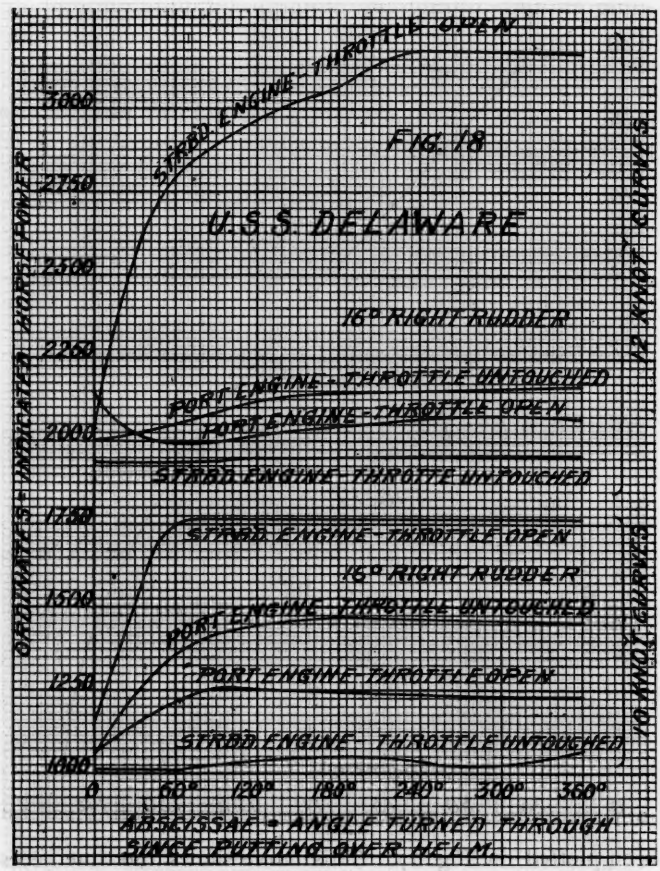


FIG. 17.

been plotted on Fig. 17. This curve shows the time necessary to bring the ship to a stop to be 1 minute 50.4 seconds. It is realized, of course, that this condition may not be entirely realized, as some reason (such as propeller vibration) which has nothing to do with the engines may make it undesirable to use this much power in backing.

TURNING.

It has always been known, of course, that when a ship turns, the inboard screw slows down if the throttle is not touched



during the turn, as is the rule ordinarily followed. In the case of an electrically-propelled ship, operating with a governor-controlled turbine, the r.p.m. of the two screws are maintained the same as they were before the turn. In order to determine exactly what effect this would produce, turning trials were carried out on the *Delaware* and the *Jupiter*. Six turns of 360 degrees were made on the *Delaware*, two at ten knots with 16 degrees right rudder, two at twelve knots with

16 degrees right rudder, two at twelve knots with 27 degrees rudder. The first four turns are shown as curves in Fig. 18. The data obtained on the last two turns is given in Table 6. The data on tactical diameter, etc., for all six turns is given in Table 7. It will be seen from Table 7 that, for speeds above ten knots the tactical diameter is practically the same, whether the turn is made with r.p.m. constant or whether the inboard screw is allowed to slow down. In the turn at 12

TABLE 6.—U. S. S. "DELAWARE," FEBRUARY 12, 1914.—TURNING TRIALS.

Turn at 12 knots speed. 27 degrees helm. Throttle untouched. Run number 5.

	I.H.P.	Pressures. H.P. valve chest.	Pressures. 1st receiver.	Pressures. 2d receiver.	Revolutions.
Starboard,	1,889	76	30	3	54
Port,	2,235	75	24	1	66
Total,	4,124				

Turn at 12 knots speed. 27 degrees helm. Maintaining same revolutions during turn. Run number 6.

	I.H.P.	Pressures. H.P. valve chest.	Pressures. 1st receiver.	Pressures. 2d receiver.	Revolutions.
Starboard,	3,469	170	60	10	66
Port,	2,047	75	22	—1	66
Total,	5,516				

Horsepower on straight run. From former records.

10 knots speed,	Starboard engine,	1,175 I.H.P.	55 turns.
	Port engine,	1,150 I.H.P.	
12 knots speed,	Starboard engine,	2,000 I.H.P.	66 turns.
	Port engine,	1,965 I.H.P.	

knots, with 16 degrees rudder and r.p.m. constant, it will be seen that the I.H.P. of the inboard screw rose steadily as the turn progressed, that the I.H.P. of the outboard screw first dropped and then rose steadily as the turn progressed. This latter peculiarity is not observed in the curves given on the 10-knot turn, probably on account of inability to indicate power frequently enough, but it is present in all of the turns

TABLE 7.—TACTICAL DATA, U. S. S. "DELAWARE."

Draught, forward, 28′ 8″ aft, 29′ 11″.

Date: February 12, 1914. Place: Off Ceiba Bank, Cuba. Depth of Water: 100 fathoms.

Serial number of run.	On straight course.			Rudder.			Engines ahead.		90 degrees.					180 degrees.				360 degrees.				
	Speed.	R.p.m. Stbd. screw.	R.p.m. Port screw.	Direction.	Angle.	Time to put over.	Starboard.	Port.	Advance.	Transfer.	Time for turning.	Rev. Starboard.	Rev. Port.	Tactical diameter.	Time for turning.	Rev. Starboard.	Rev. Port.	Final Diameter.	Time for turning.	Final speed on turn.	Final drift angle measured at foremast.
					°	″					′ ″				′ ″				′ ″		
1	10	55	55	Right	16	6	Throttle untouched		700	425	3-07	50	55	890	5-50	48	55	890	11-43	7 2	4½°
2	10	55	55	do.	16	6	Revs. of both maint'd.		750	495	2-19	55	55	1010	6-04	55	55	1010	11-58	7.8	1½°
3	12	66	66	do.	16	6	Throttle untouched		750	460	3-37	60	66	940	4-57	59	66	940	9-57	9.0	4½°
4	12	66	66	do.	16	6	Revs. of both maint'd.		810	490	2-54	66	66	980	5-10	66	66	980	10-04	9.75	2°
5	12	66	66	do.	27½	6	Throttle untouched		555	350	2-08	No	data	710	4-01	54	66	710	8-26	7.2	5°
6	12	66	66	do.	27	6	Revs. of both maint'd.		600	340	2-00	do.		710	3-53	66	66	710	7-58	8.9	4½°

Notes on wind and sea.—Wind E.N.E., force 1; sea smooth.

made by the *Jupiter*, as will be seen later on. The reason for the shape of the horsepower curves when turning is that the inboard screw is maintaining a constant r.p.m. at a much lower virtual speed of the ship than these r.p.m. would give if driving ahead, so, of course, the power goes up; the condition is still more exaggerated as the ship slows during the turn. The outboard screw, at the beginning of the turn, is maintaining a constant r.p.m. at a considerably higher virtual speed of the ship than these r.p.m. would give if driving ahead, so the power required drops; but as the ship slows during the turn a point is soon reached where this excess speed is lost, and as the speed of the ship falls lower the power commences to rise. From Table 5, on the turn at 12 knots with 27 degrees rudder, we see that the greatest increase of power on the inboard screw was 73.5 per cent. of the normal driving power; the greatest increase on the outboard screw was 4.2 per cent. of the normal driving power, and the total increase of power was 39 per cent. of the total driving power.

In Figs. 19 and 20 are shown the results of turning trials on the *Jupiter*. Two turns were made at 12 knots and two at 14 knots; one turn was made to starboard and one to port in each case; the turn was made through 180 degrees in each case using 25 degrees rudder. It was possible to get very accurate results on these trials as the r.p.m. were maintained exactly constant by the governor, and horsepower readings were taken every five seconds. The curves obtained are all similar to those shown on the *Delaware's* 12-knot turn in Fig. 18. The reason for the shape of these curves has already been given under the explanation of the *Delaware's* curves. The difference in the curves obtained when turning to starboard and port are due, partly to the fact that the rudder angles were probably not exactly the same, and partly to the difference in the effect of wind and sea on the two sides. The greatest increase of power occurred on the 14-knot turn to port. The inboard screw increased in power 53.5 per cent., the outboard screw increased 4.5 per cent., and the total increase of power was 29 per cent. These percentages are con-

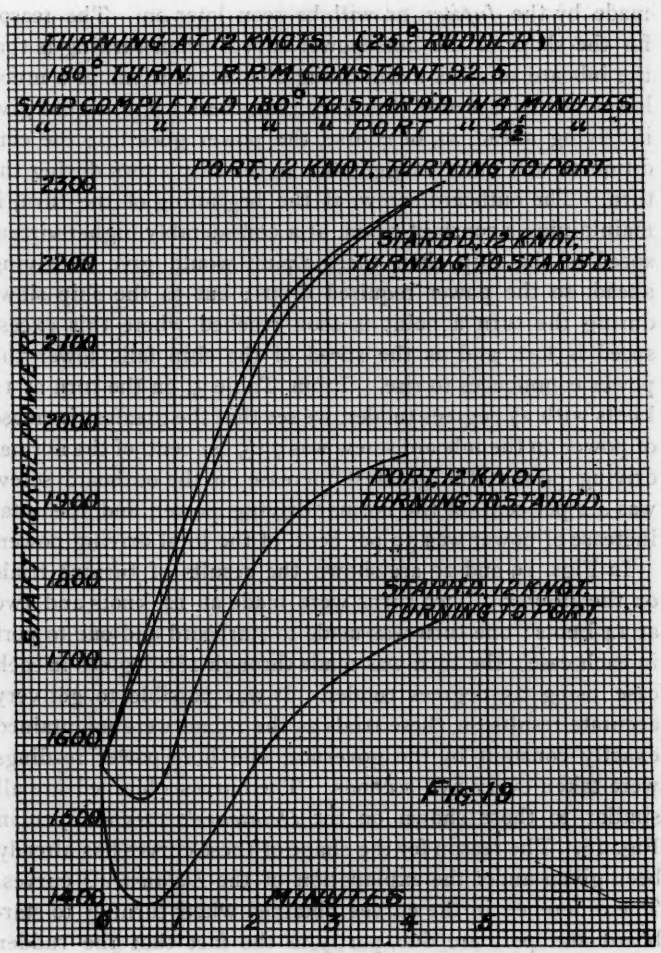


Fig. 19

siderably lower than those obtained on the *Delaware*, but the *Delaware's* rudder is 50 per cent. larger than that of the *Jupiter*, so that she turns and slows faster than the *Jupiter* and consequently takes a larger increase of power.

The effect of turning on an electrically-propelled ship would

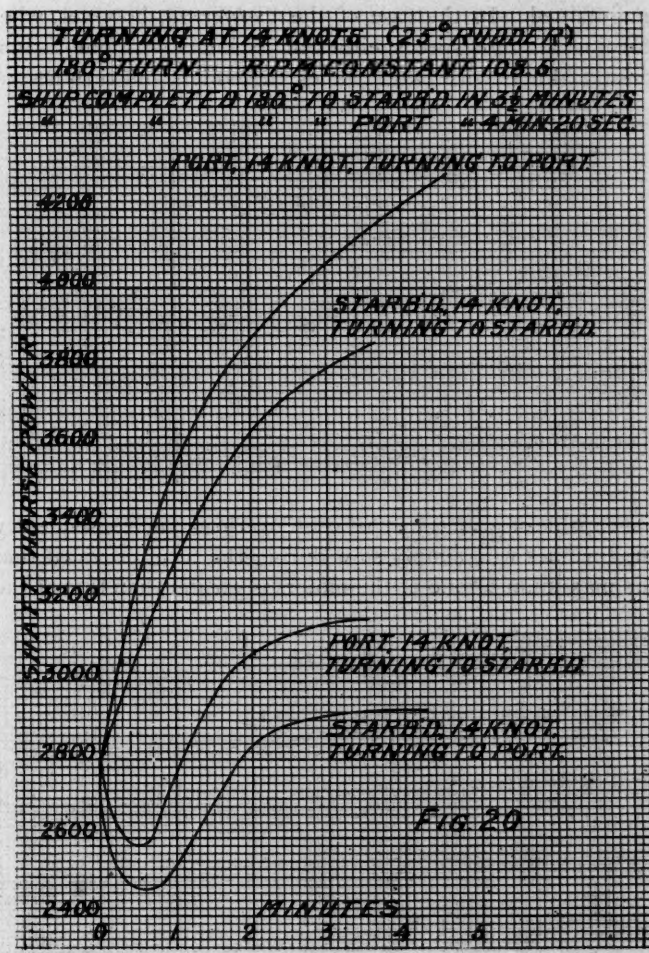


FIG. 20

depend on the design of the motors and the turbines. In the case of the *Jupiter* the power plant of the ship is sufficient to maintain constant r.p.m. at all speeds during a turn; also the maximum torque of the motors is sufficient to insure that they will stay in step during a turn if the proper excitation is

maintained. In the case of the *California*, however, when turning at high speeds, either at 18 knots with one turbine or at 21 knots with two turbines, the boilers and turbines are

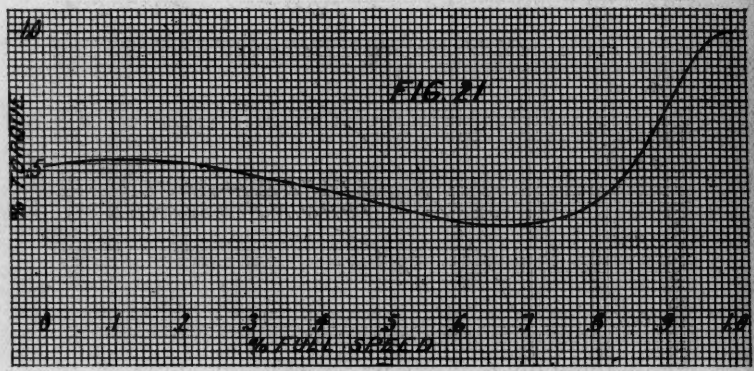


FIG. 21

not capable of giving a 39 per cent. overload, and consequently the turbines will simply slow during the turn, provided that the maximum torque of the motors is greater than that of the

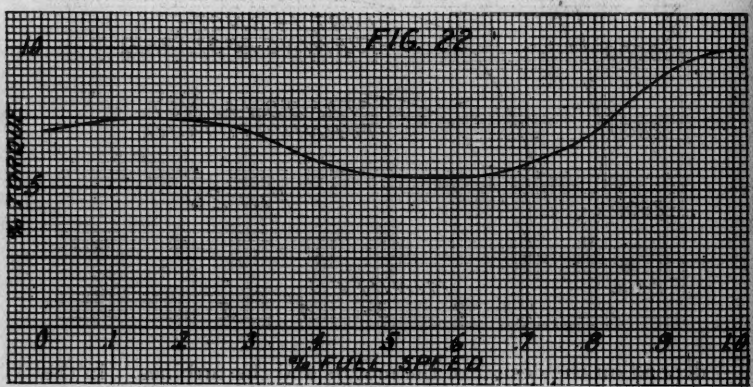
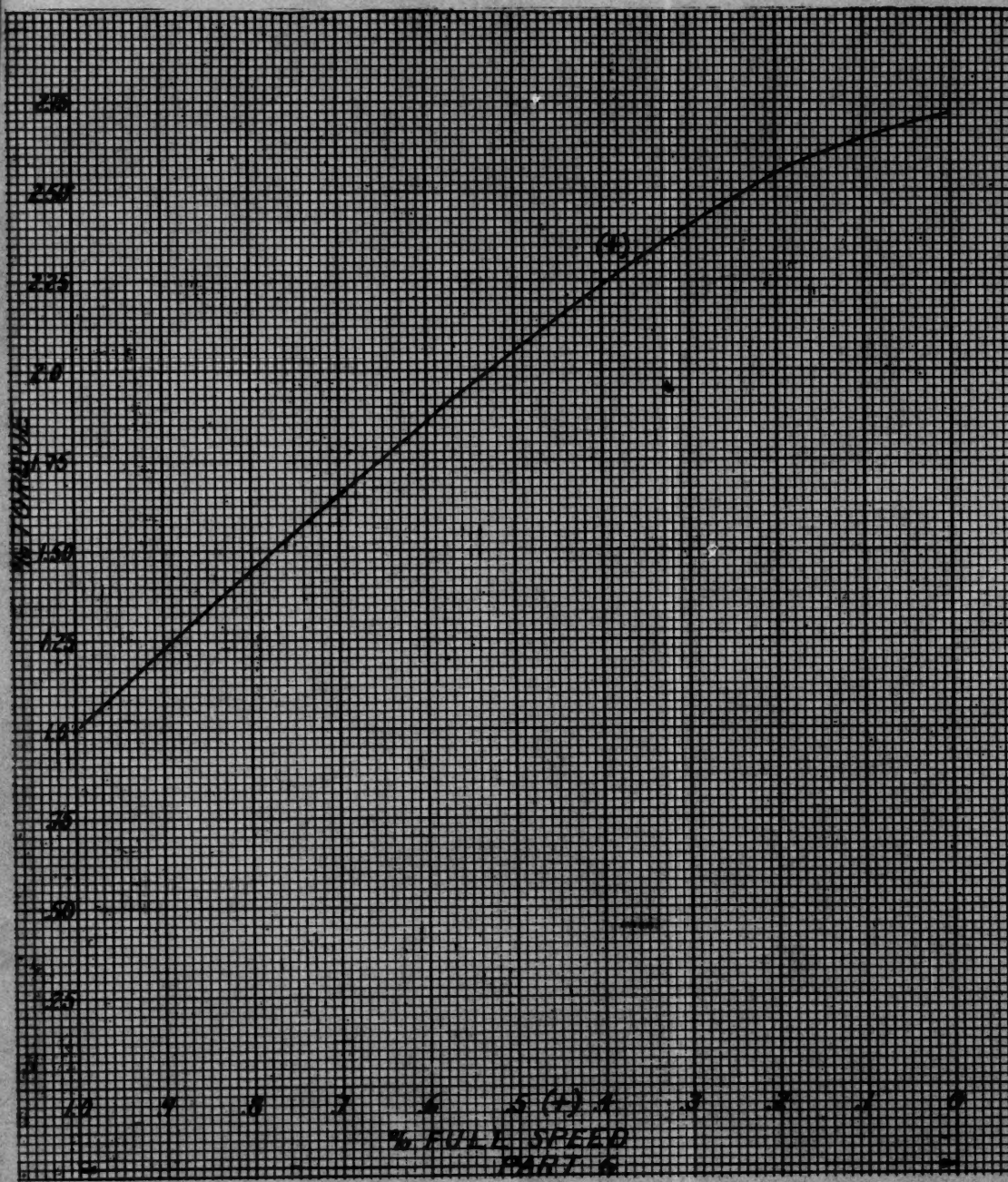
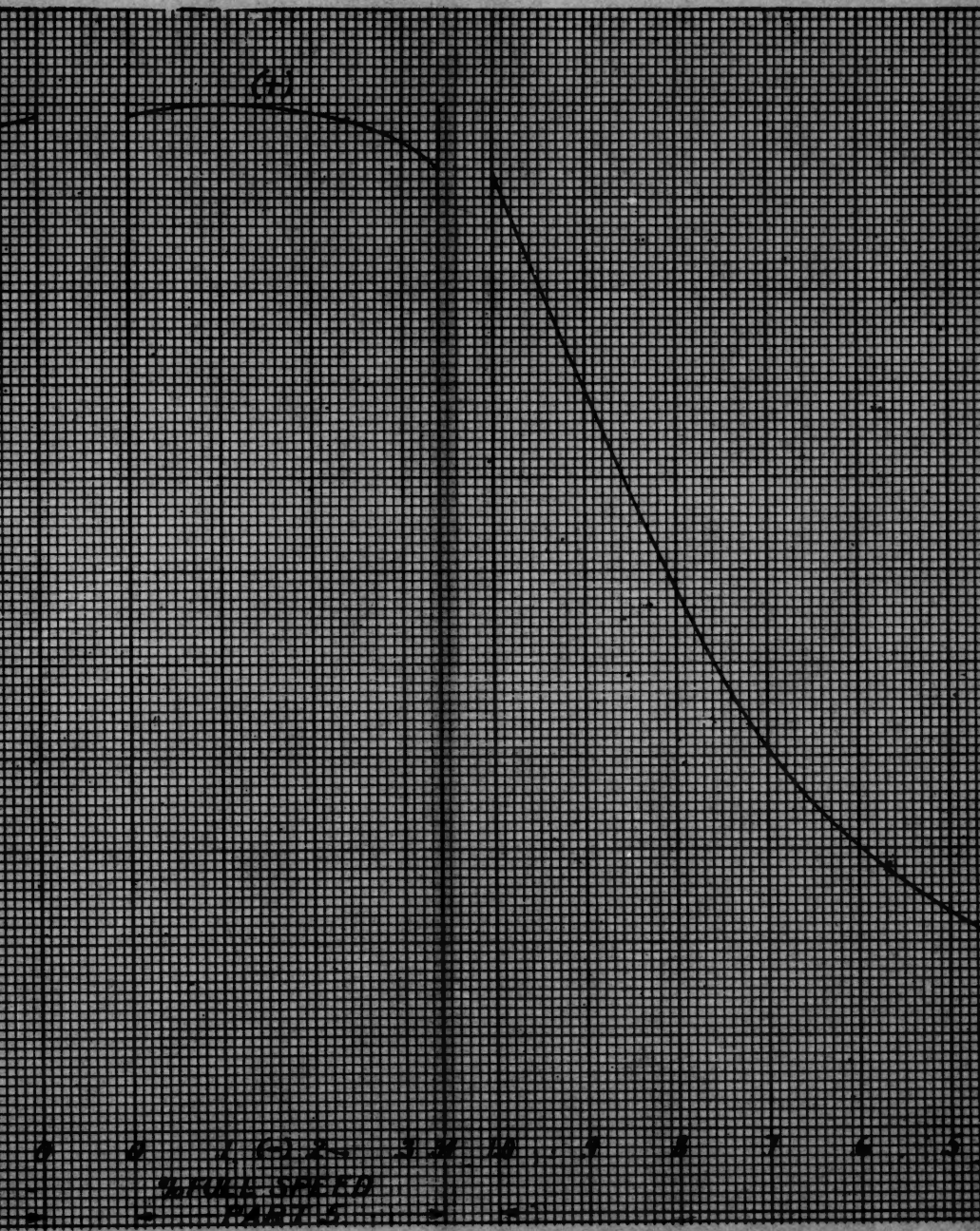


FIG. 22

turbine. In other words, the *California* will turn without any reduction of r.p.m. at all low speeds, say below 16 knots, and will slow her r.p.m. during the turn at all speeds above this.


250
225
200
175
150
125
100
75
50
25
0
% FULL SPEED
PART 6


(E)
STALL SPEED

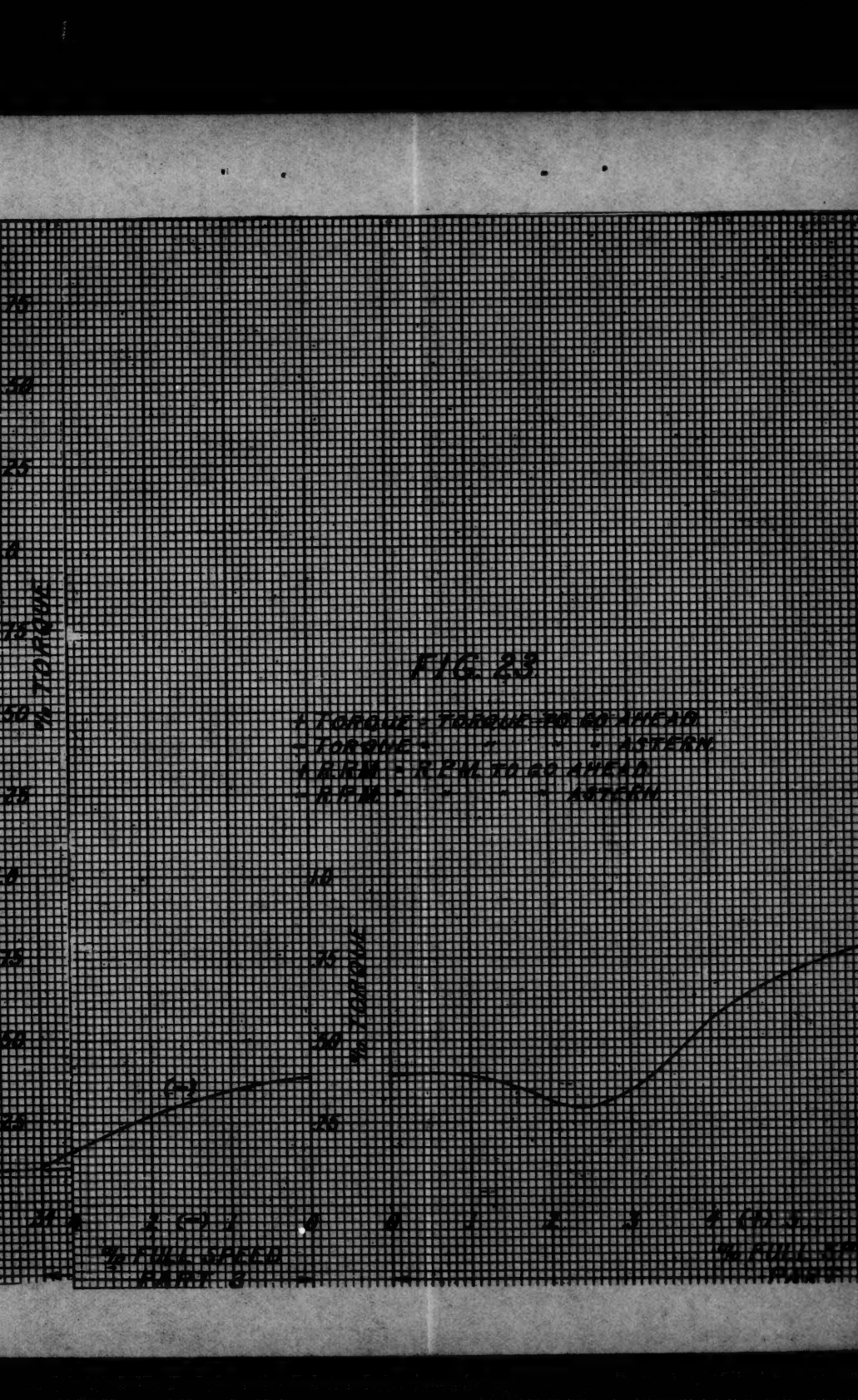
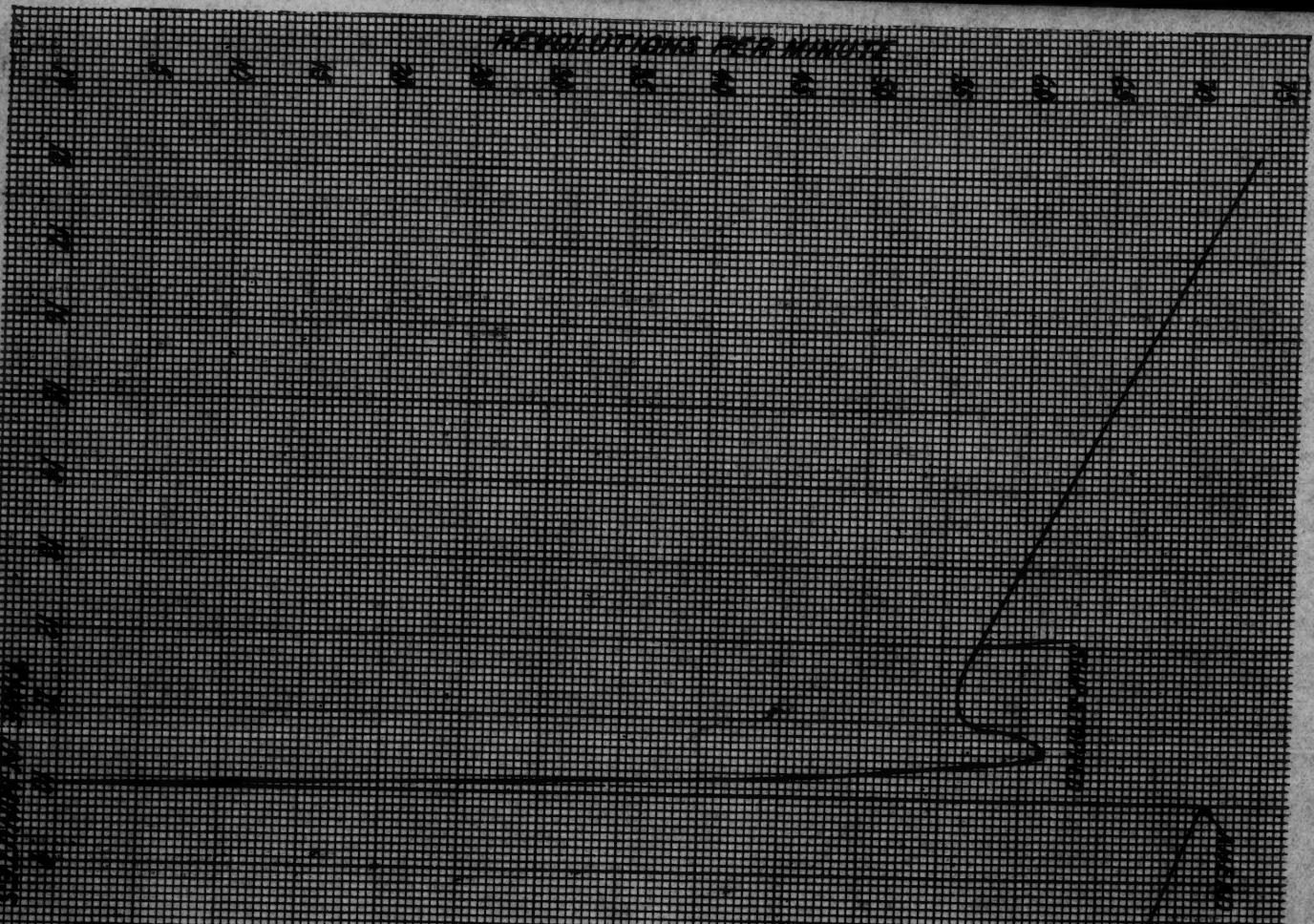


FIG. 23
Torque - Torque to go ahead
Torque - " " astern
KLCM - KLCM to go ahead
KLCM - " " astern
% Full speed ahead astern

10
75
50
25
EFFICIENCY %
FULL SPEED
POWER


REVOLUTIONS PER MINUTE

SHIP STOPPED
REVOLUTIONS PER MINUTE
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
8
7
6
5
4
3
2
1
0

FIG. 24.

AEROPLANE POWER PLANTS.

By Lieutenant Warren G. Child, U. S. N., Member.

By the term "power plant" is meant those parts of the aeroplane which furnish its motive power—motor, starting device, radiator, propeller, fuel pump and fuel tanks.

All aeroplane motors now in use are single-acting. All use gasoline for fuel. Most aeroplane motors are of the four-cycle type, though a few operate on the two-cycle principle.

In the four-cycle, or Otto cycle motor, an impulse is given to each piston once in two revolutions, or in four strokes. The operation, or cycle, is as follows: During the first, or *induction*, stroke the mixture of gasoline vapor and air is drawn into the cylinder, through the open inlet valve, by the suction of the downward-moving piston: During the second, or *compression*, stroke, inlet and exhaust valves being closed, the charge is compressed by the ascending piston into the clearance space between the piston, on top center, and the cylinder head. Just before the end of the compression stroke the charge is ignited by an electric spark and the expanding gases force the piston downward, thus rotating the crank shaft, inlet and exhaust valves remaining closed, this operation being called the *power* stroke. Just before the end of the power stroke the exhaust valve is opened, and on the fourth, or *exhaust*, stroke the burned gases are driven from the cylinder by the ascending piston. The cycle is then repeated.

In the two-cycle motor an impulse is given to each piston during each revolution, or every two strokes. There is first the power stroke, in which the piston is driven downward by the expanding gases, valves being closed, and the next charge is being slightly compressed in the crank case. Just before the end

of this stroke the exhaust valve opens, allowing the burned gases to escape. A little later the inlet valve opens and a new charge enters the cylinder, the exhaust valve closing about the same time. After the piston has moved a short distance on the up stroke, or compression stroke, the inlet valve closes, the charge is compressed and, just before the end of the stroke, ignited. In this cycle there is more or less mixing of the burned gases and the incoming charge, with a consequent loss in power. With the two-cycle type of motor lubrication is effected by mixing lubricating oil with the gasoline. Compression is never as high as in the four-cycle type.

The greater the number of cylinders in a motor the greater the number of impulses per revolution, of course, and the more uniform the turning moment. However, the complications added by the multiplicity of cylinders may offset these advantages.

Other things being equal, the weight per horsepower of a single motor is always less than that of two or more units of the same total horsepower.

In comparing aeroplane motors the most important factors to consider are fuel and oil consumption, weight, reliability, durability, accessibility and simplicity—probably in the order named. A true comparison can be made only by considering the weight, due solely to the power plant, which the aeroplane must carry on a flight of a certain number of hours. This is the weight of the complete power plant, with fuel and oil, for the duration of the flight; it is termed the gross weight. In such a comparison the great importance of fuel economy is evident; it may even outweigh the advantages of light weight in a light but uneconomical motor. For example, the fuel and oil consumption of a well designed four-cycle motor is about .53 pound per horsepower hour, while the total weight carried, or the gross weight, is about 11½ pounds per horsepower for a ten-hour flight, whereas the corresponding fuel and oil consumption and weight per horsepower of a light rotary motor are about .9 pound and 14 pounds.

Weight of power plant is next in importance to fuel economy. In fact, the two are closely inter-related, since the higher the fuel economy the less the gross weight. Light motor weight is a matter of design, materials used and quality of workmanship. In the last year or two aluminum alloys have played a most important part in weight reduction. They are used successfully for crank cases, cylinders, pistons, and for many minor castings. The elements alloyed with aluminum are copper, zinc, tin, magnesium, nickel, tungsten, chromium and antimony. The alloys vary in specific gravity from 2.5 to 2.9. Unfortunately, many manufacturers of aeroplane motors have obtained lightness at the expense of reliability and durability.

No motor, however well designed, will stand up under the hard usage in service unless the metals used in its construction are of the best, properly heat-treated, and the workmanship is of the highest order.

The question is often asked: "Why do not aeroplane motors stand up as well as automobile motors?" There are two principal reasons: the weight per horsepower of the automobile motor is much greater than of the aeroplane motor, and the automobile motor seldom operates at full load, whereas the aeroplane motor is made as light as possible (about one-third the weight of the automobile motor) and, in service, is run practically all the time at close to full power.

In order to show the requirements of naval aeroplane motors, the following is taken from a recent proposal for bids:

GENERAL SPECIFICATIONS FOR MOTORS FOR AEROPLANES.

All parts to be of the best material and workmanship. Engine to deliver between 140 and 160 brake horsepower with muffler attached. They shall be well balanced and produce no excessive vibration at any power. To be capable of being throttled down to 20 per cent. of the revolutions per minute for full power. The weight of the engine complete,

with ignition system, magnetos, carburetors, pumps, radiator, cooling water and propeller, not to exceed 5 pounds per brake horsepower. Engine to be fitted with some type of compression release as a means of stopping it. To be fitted with a practical means of starting from pilot's seat when installed in an aeroplane. All moving parts not lubricated by a splash or forced-lubrication system to be readily accessible for inspection, adjustment and oiling. Ready means shall be provided for checking and making adjustment to the timing of the engine. All parts to be machined all over where possible. Fillets to be of ample radius to insure strength. To have an accurate and positive lubricating system which will insure a uniform consumption of lubricating oil proportional to the speed of the engine. All parts subject to corrosion to be protected from the effects of salt water. To be fitted with an approved attachment for obtaining the revolutions per minute. To be provided with means for preventing fire in case the engine is turned upside down. A hand throttle lever and connections to carburetor to be provided that can be applied for convenient operation by the pilot. This lever to be designed with a positive means of retaining it at the throttle adjustment desired by the pilot. All bolts and screws, without any exception, to be provided with an approved positive means for preventing backing out due to vibration. No soft solder to be used in any part of the power plant.

Acceptance tests:

1. To determine power revolution curve, and fuel and lubricating-oil economy.

2. Three runs each of 8 hours' duration at full power.

3. Eight separate and distinct runs in the following manner: Engine to be started by the use of the starting gear; to be run for the first 5 minutes at 95 per cent. of full power while inclined at an angle of 15 degrees to represent climbing; then to be run for 22 minutes at loads varying between 75 per cent. and 95 per cent. of full power; then to be throttled down to 20 per cent. of full power revolutions per minute, and the

engine to be tipped 30 degrees to simulate gliding, and to run in this position for 2 minutes; then to be leveled and run at full power for 1 minute. Engine to be then stopped, and after 5 minutes a similar test to be repeated, and this shall be continued until 8 separate runs have been made.

Note.—No adjustments or alterations shall be made during any of the tests of intermissions. The failure to complete any test in a satisfactory manner shall require that that set of tests be made again.

During all tests an accurate record shall be kept of the following:

Total number of revolutions for each test.
Revolutions per minute for each test.
Lubricating-oil consumption for each test.
Gasoline consumption for each test.
A record of thrusts for test 2.

At the successful completion of these tests the engines shall be broken down and inspected. No parts shall show undue wear or deterioration, and the weights and balance of the separate parts shall be in conformity with the specifications.

SPECIFICATIONS FOR PARTS.

Lower crank case.—To have an oil capacity for 5 hours running at full power, or fitted with an efficient automatic device to keep constant supply of oil at pump suction. If oil is supplied from a tank, tank and oil pump are to be provided. To be fitted with a sight gage to show the height of the lubricating oil. Oil system to have a pressure gage capable of being mounted on an instrument board, and means to turn this system on and off at will. To be fitted with drain plugs. If possible, to be capable of removing from engine with engine installed on engine beds. To be fitted to prevent oil leaks at joints and connections. All fillets to be of ample radius to insure strength.

Upper crank case.—To be fitted with an efficient system of

relieving pressure in the crank case. Wherever possible, when webs are fitted which can not be machined they should be of uniform thickness. Wherever attachments are made to the upper crank case they should be sufficiently reinforced to insure strength. All fillets to be of ample radius to insure strength.

Cylinders.—To be fitted to take two spark plugs. To be cooled to prevent excessive heating of valves. Cylinders to be counterbored to prevent piston wearing a shoulder at either end of the stroke. They shall be secured to the crank case so as to prevent oil leaking. To be fitted with an efficient detachable muffling system. To be attached to the crank case with sufficient safety factors to insure against cylinders blowing off. Cylinder holding-down bolts to be subjected to a uniform load on all members. Cylinders to be machined on the outside as well as inside. Spark plugs to be accessible and removable without removing any engine parts. Spark-plug points to extend into the combustion chamber. Any pockets in the head of the cylinder where burnt gas might collect to be avoided. Water jacket, if used, to be a noncorrosive metal.

Pistons.—To be balanced and finished so that homologous parts shall be of uniform thickness. Sufficient thickness shall remain after the piston-ring grooves have been cut to insure ample strength. Means shall be taken to prevent an excess of oil entering the combustion space. Wrist-pin bearings to be bushed with bronze. Means to be provided to prevent wrist pin or bushings from touching the cylinder walls. All pistons to be of the same weight and to balance at the same point.

Piston rings.—To be preferably of the leak-proof type, made in two parts, and not less than ¼ inch wide.

Connecting rods and bearings.—They shall be machined all over and balanced with liners (I-beam cross section, preferred), crank-pin bearing cap and crank-pin bolts and nuts in place. All connecting rods to be of the same weight and to balance at the same point. Crank-pin bolt heads to be counter-

sunk into connecting rod to prevent turning. Ends of wrist pin to be beveled.

Crank shaft and main bearings.—Crank shaft to be carefully balanced and to be machined all over. If the crank shaft is made with a flange, to which it is the intention to secure the propeller, this flange shall be at least ½ inch thick. Special attention should be given to the strength of the crank webs.

Cam shaft.—To be machined all over and balanced. Cams to be integral with the shaft, case-hardened to prevent pounding out. To be fitted with some means to decrease the amount of wear and friction on the cams and on the end of the push rods.

Push rods, valve gear, rocker arms, etc.—To be machined all over. To have an accurate and positive means of opening and closing valves. Clearance between valve stems and actuating mechanism of both intake and exhaust valves to be capable of adjustment. All parts to have sufficient strength factor to insure against distortion, such as bending of rocker arms and push rods, if employed. Rocker arms to be machined all over.

Intake manifold and carburetor.—To be smoothly finished on the inside. No bolts or lugs to pass through or extend into the inside of the intake manifold. Shaft turns or bends to be avoided and the distance from carburetor to any intake port to be approximately the same. Means to be provided for heating the mixture *after* entering the intake manifold. The use of two carburetors is preferred. The carburetor to be so located as to provide for the minimum amount of vibration (*i. e.*, as near the line of the crank shaft as possible). The face of cylinder to which the intake manifold is secured to correspond exactly to the cross section of the air-manifold flange. Means shall be provided for making accurate and permanent adjustments to the carburetor, unless automatic in action, and means to prevent these adjustments changing due to vibration. Means for adjustments from pilot's seat shall be provided to care for changes in altitude.

Muffler.—To be suited to the motor, and, if practicable, to be fitted with a cut-out.

Magnetos.—Two magnetos to be used, either one capable of developing 90 per cent. of full power. To be waterproof. Positive means shall be provided for securing magneto leads to the distributor.

Leads to spark plugs to be waterproof.

Angle of advance to be controlled from pilot's seat.

Circulating pump.—To have a capacity sufficient to cool the engine when running at full power in air of 72 degrees F.

Propeller.—To be designed for a speed of 75 miles per hour at normal revolutions of the motor, to be accurately balanced, and to have an efficiency of at least 70 per cent. To have no metal tips, and to be not greater than 8 feet in diameter.

POWER PLANT PARTS.

Starting device.—Motors up to about 140 horsepower are usually started from the pilot's seat by means of a crank operated by hand. On motors above 140 horsepower a power starter is necessary.

One make of starter, in successful operation, uses compressed air mixed with gasoline. This installation, with compressor and air tank, weighs about 40 pounds.

With a specially designed light battery it should be perfectly feasible to use a combined electric starting and ignition system on aeroplanes, as now used in automobiles. The generator of such a system could also be used for a wireless equipment.

Radiators.—Radiators are of the same general types as those used in automobiles, though the shapes are, in many cases, different. They are designed to suit the motor and are located in various places—above the motor, or at the sides or the fuselage, at its nose, or under it.

A radiator should be as light as possible and should be of such design as will cut the head resistance to a minimum.

Propellers are of wood, spruce, birch, maple, white oak or walnut, well varnished. Many of them have the ends protected by a sheathing of thin copper or brass. It is of the utmost importance that this sheathing be well secured and the propeller perfectly balanced. In one instance, during a flight, the sheathing flew off both ends of the propeller, splitting both blades to the hub, tearing the engine from the foundations, carrying away a section of the tail bamboo and rupturing a number of stays.

The propeller may be directly driven by the crank shaft, from the cam shaft at one-half motor speed, from a separately-geared tail shaft at some intermediate speed, or by chain and sprocket drive.

A propeller should be designed to meet the requirements of the machine as to diameter and speed in miles per hour at normal speed of the motor. It should have an efficiency of at least 70 per cent. Propeller speeds range from about 900 to about 1,500 r.p.m.

Fuel supply.—The main supply of fuel is carried in tanks, fitted with swash plates, in the body of the machine. The immediate supply is usually carried in a small tank, of streamline form, above the motor, called the service tank. A great danger exists in the location of the service tank, in that the piercing of the tank by a bullet or otherwise might cause a flow of gasoline upon the motor exhaust thereby causing a fire. This danger is partly eliminated by locating the tank as far from the exhaust as possible.

The service tank may be eliminated, with a consequent reduction of weight and head resistance, if either automatic gravity tank or the pressure system is used, but at the disadvantage of having no reserve supply of fuel in case the system becomes inoperative from a loose connection, by the piercing of the tank, by breakdown of mechanism, or from any other cause. Taken all in all, however, the advantage appears to rest with the automatic gravity tank.

Probably the best arrangement for supplying fuel would be

a fuel pump, operated by the motor, to supply gasoline directly to a small tank near the carburetor. Such a pump system should be developed.

Fuel is pumped from the main tank to the service tank, which is usually of sufficient capacity for a one-half hour supply, by means of a wind-mill type of pump; that is, a pump mounted in an open position on the aeroplane and driven by a fan propelled by the force of the air against its blades in flight. The fuel flow to the carburetor is by gravity. On machines of short cruising radius, as school machines, the whole supply of fuel is carried in a large service tank.

On some machines the service tank is eliminated and the main tank kept under pressure, pressure being supplied by an engine-driven air pump.

POWER AND EFFICIENCY.

Our unit of work is the foot-pound, which represents the work done in lifting one pound through a distance of one foot.

The unit of power is the horsepower which represents work done at the rate of 33,000 foot-pounds per minute.

The work done on the explosion stroke in a motor is equal to the area of the piston multiplied by the mean effective pressure developed during the stroke, acting through a distance equal to the length of the stroke.

Since in a four-cycle motor there is only explosion in every two revolutions, the work done per cylinder during each explosion becomes

$$\pi r^2 \times m.e.p. \times \frac{L}{12} \times \frac{n}{2},$$

where n = r.p.m. and L = length of stroke in inches and r = cylinder radius, or ½ diameter.

The work done per cycle in a motor of M cylinders then equals the above equation × M.

Since one horsepower is work performed at the rate of 33,000 foot-pounds per minute, the indicated horsepower of a four-cycle motor =

$$\frac{\pi r^2 \times m.e.p. \times L \times n \times M}{33{,}000 \times 12 \times 2}.$$

For a two-cycle motor this, of course, is doubled, since there is an explosion every revolution.

Owing to the great difficulty of obtaining accurate indicator cards of a high-speed motor such motors are rated according to their brake horsepower, which is the power delivered to the shaft, and is equal to the I.H.P. minus the mechanical friction. It may be measured by any one of several different types of dynamometers—electric, water brake or fan.

$$\text{B.H.P.} = \frac{2\pi \times \text{r.p.m.} \times \text{scale reading} \times \text{length of arm}}{33{,}000}.$$

With an arm 63.025 inches long this formula reduces to

$$\text{B.H.P.} = \frac{r.p.m. \times \text{scale reading}}{1{,}000}.$$

Suppose it is desired to ascertain the H.P. required to maintain a 6,000-pound aeroplane in horizontal flight at 60 miles per hour

W = weight of machine = 6,000 pounds.

Head resistance $= \frac{W}{6}$ (approximately) = 1,000 pounds.

Effective horsepower $= \frac{1{,}000 \times 60 \times 5{,}280}{60 \times 33{,}000} = 160.$

Assuming a propeller efficiency of .75 the motor must deliver $\frac{160}{.75} = 213$ H.P. to maintain the machine in horizontal flight at this speed.

Any excess power is available for other purposes. The rate of climb with two 160-B.H.P. motors would be as follows:

$320 - 213 = 107$ excess horsepower;

Propeller efficiency = .75;

$107 \times .75 = 80.25$ available excess horsepower;

$\frac{80.25 \times 33{,}000}{6{,}000} = 441$ feet per minute climb.

THERMAL EFFICIENCY.

This is the ratio of the heat utilized to the heat available.

One pound of gasoline has a calorific value of from 18,000 to 21,000 British thermal units. The work equivalent of one B. t.u. is 778 foot-pounds. That is, 778 foot pounds of work may be done by the dissipation of one B.t.u. Allowing 19,000 B.t.u. per pound of gasoline, the amount of work which may be done by one pound, if all the heat is utilized is

$$19{,}000 \times 778 = 14{,}782{,}000 \text{ foot-pounds.}$$

Suppose that the fuel consumption of a motor is .7 pound per H.P. per hour. One H.P. represents

$33{,}000 \times 60 = 1{,}980{,}000$ foot-pounds of work per hour. Therefore, the heat in .7 pound of gasoline is capable of producing

$$7 \times \frac{14{,}782{,}000}{1{,}980{,}000} = 523 \text{ H.P. per hour.}$$

But, since from this fuel only one horsepower is obtained, the motor has a thermal efficiency of only $\frac{1}{5.23} = 19$ per cent.

The ratio of air to gasoline, by weight, after carburetion, varies from 11 to 1, for the richest mixture, to 18 or 19 to 1, for the leanest mixture, and 12 or 15 to 1 ordinarily.

Maximum thermal efficiency is obtained with a lean mixture and maximum power with a rich mixture.

VOLUMETRIC EFFICIENCY.

This is the ratio of the volume of charge taken in per suction stroke, at normal temperature and pressure, to the piston displacement.

Piston displacement is the space swept through by the piston in passing from one dead-center to the opposite dead-center; that is, through one stroke.

The amount of gas, at atmospheric temperature and pressure,

drawn into a cylinder during the suction stroke is never equal to the piston displacement. It is dependent on the motor speed —decreasing as the speed increases, on the size and amount of opening of the inlet valve, on the temperature of the cylinder walls, on the valve timing, on the friction of the passages from the carburetor to the cylinders and on the suction power of the piston.

Suppose a mixture contains W pounds of air to 1 pound of gasoline.

Let a = lbs. of gasoline per B.H.P. per hour;
b = B.H.P. per cylinder.

Then each cylinder receives $ab\ (W + 1)$ pounds of mixture per hour.

Since 1 lb. of air at 70 degrees $\times$ 13.342 cu. ft., and
1 lb. of gasoline at 70 degrees = 4 cu. ft.,

this mixture has a volume of $ab\ (13.342 + 4)$ cu. ft. per hour.

Let s = piston speed in feet per minute;
a = piston area in sq. ft.

Then $\frac{sa}{4}$ = volume of suction in cu. ft. per minute (since there is only one suction in four strokes), and $60 \times \frac{sa}{4}$ or $15\ sa$ = piston displacement in cubic feet per hour during suction strokes.

Since the volumetric efficiency is the ratio of the mixture volume to the piston displacement, we have

$$\text{Volumetric efficiency} = ab\ \frac{(13.342\ W + 4).}{15\ sa}$$

TYPES AND CLASSES.

Aeroplane motors may be divided into two general types: (1) air cooled, in which cooling is effected by the circulation of air around the cylinders, either by the rotation of the motor

itself, or with stationary cylinders, by means of a blower operated by the motor (the operation of such a blower requires about 10 per cent. of the horsepower developed), and (2) water cooled, in which cooling is effected by the circulation of water driven by a pump through jackets surrounding the cylinders.

In the air-cooled motor radiation is assisted by ribs on the cylinders. In the water-cooled motor a radiator is necessary to remove the heat from the cooling water. About 15 per cent. of the B.t.u. generated in the cylinders must be removed by the cooling system.

One pound of head resistance is equivalent to about ten pounds of load, and one square foot of head resistance to about 80 pounds of load. The great importance of proper radiator design is thus apparent. In a recent test in the wind tunnel one radiator showed a resistance of 5.7 pounds at 50 miles per hour, while another of a more common type, and of only slightly greater cooling efficiency, showed 22.8 pounds, yet the latter type is in much more general service than the former.

Motors are divided into classes according to the disposition of cylinders, as follows:

Vertical.—4, 6 or 8 cylinders in a vertical line, as the Sturtevant, Hall-Scott, Roberts and Wisconsin.

Diagonal.—"V" — 4 or more cylinders in two rows, inclined at 60 degrees or 90 degrees to each other, as the Thomas, Van Blerck, Kemp, Rausenberger, Sturtevant and Curtiss.

Horizontal.—4 or more cylinders on a side, directly opposed, as the Ashmusen.

Rotary.—5 or more cylinders, usually an odd number, set radially upon the case and revolving about the crank shaft which is stationary, as the Gyro.

Radial.—3 or more stationary cylinders set radially about the crank shaft. (No American example.)

Of these classes the vertical of 4 and 6 cylinders, and the diagonal of 8 and 12 cylinders, are the most common.

The following descriptions will illustrate the various classes of American-made motors:

ROBERTS MODEL 6X MOTOR.

This is a six-cylinder, vertical, two-cycle, motor rated at 75 H.P. at 1,100 r.p.m.; bore 4½ inches; stroke 5 per cent.; weight 240 pounds.

Cylinders are cast separately of a special aluminum alloy called "Aerolite," which has a specific gravity of 2.73 and a tensile strength of 38,000 pounds per square inch, and contains neither zinc nor copper.

Back firing in this motor is prevented by a by-pass, consisting of a series of alternate flat and corrugated plates about three inches long, which entirely fill the passage from the base to the cylinder and divide the transfer port into a large number of small passages, or cells. These cells have such a large surface compared to the area of the passage through them that should a flame start downward through them it is cooled and extinguished. This by-pass also assists to vaporize the mixture.

The crank case is made from an alloy of aluminum and magnesium, called magnalium.

The crank shaft is drop-forged, double-heat treated, and is bored hollow in both the shaft proper and the crank pins. The journals are especially long to insure proper bearing surface and lubrication.

The connecting rods are I-beam section, drop-forged from crucible vanadium steel. The bearings in the lower end of the connecting rod are babbitt-faced bronze, and in the piston phosphor-bronze running on a hardened-steel pin secured in the piston.

Ignition is by means of a Bosch magneto so arranged as to secure the maximum spark at any point of advance, accom-

plished by rotating the armature and thus changing its setting instead of rotating the breaker. The mixture of gas and air passing from the carburetor to the base of the motor is controlled by means of a rotating tubular distributor, which revolves in a bronze-bushed housing at the same speed as the motor. This distributor regulates the admission of the mixture to the base by opening and closing ports cut in the wall of the crank case. From the carburetor the gas enters the revolving tube at the center and thence passes to each one of the base compartments in turn through openings in the wall of the tube. Two Kingston carburetors are used.

All gears are encased and run in oil.

Cooling is effected by means of a large-capacity gear-type pump driven off the main shaft.

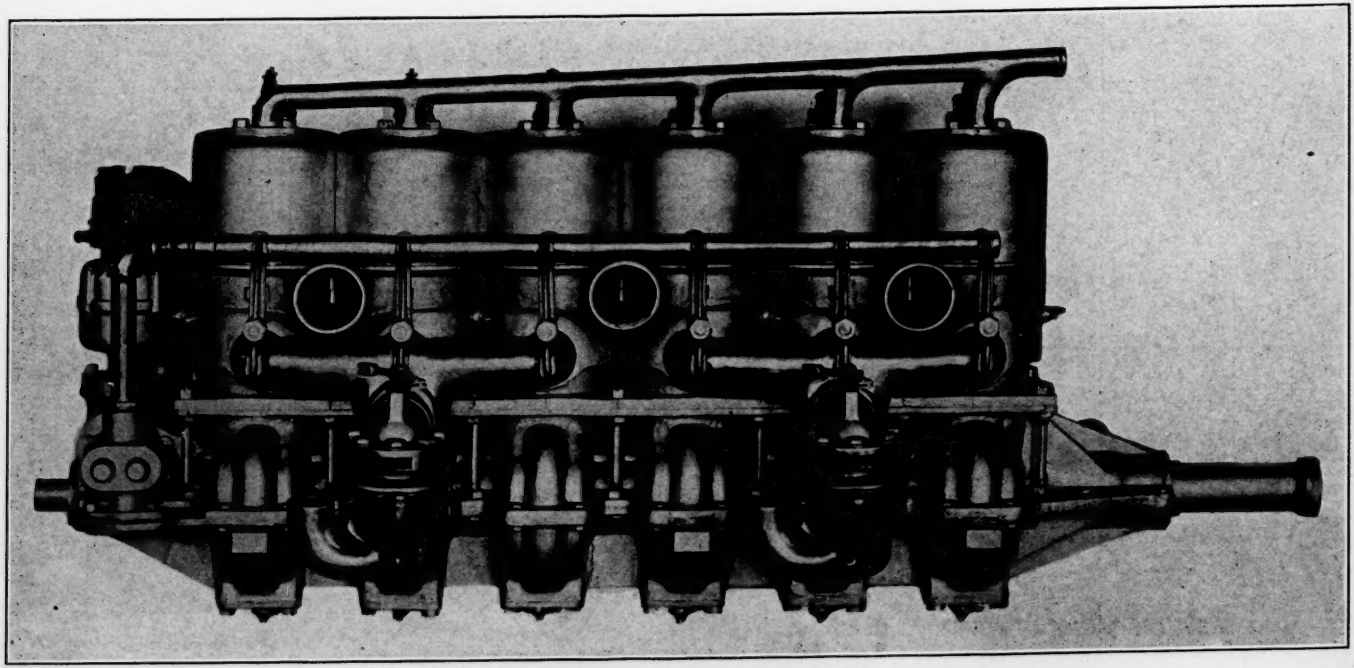
HALL-SCOTT MODEL A-5 MOTOR.

This is a vertical, six-cylinder, four-cycle motor; bore 4 inches; stroke 5 inches; guaranteed horsepower 125 at 1,300 r.p.m.; weight 660 pounds with radiator, cooling water and propeller.

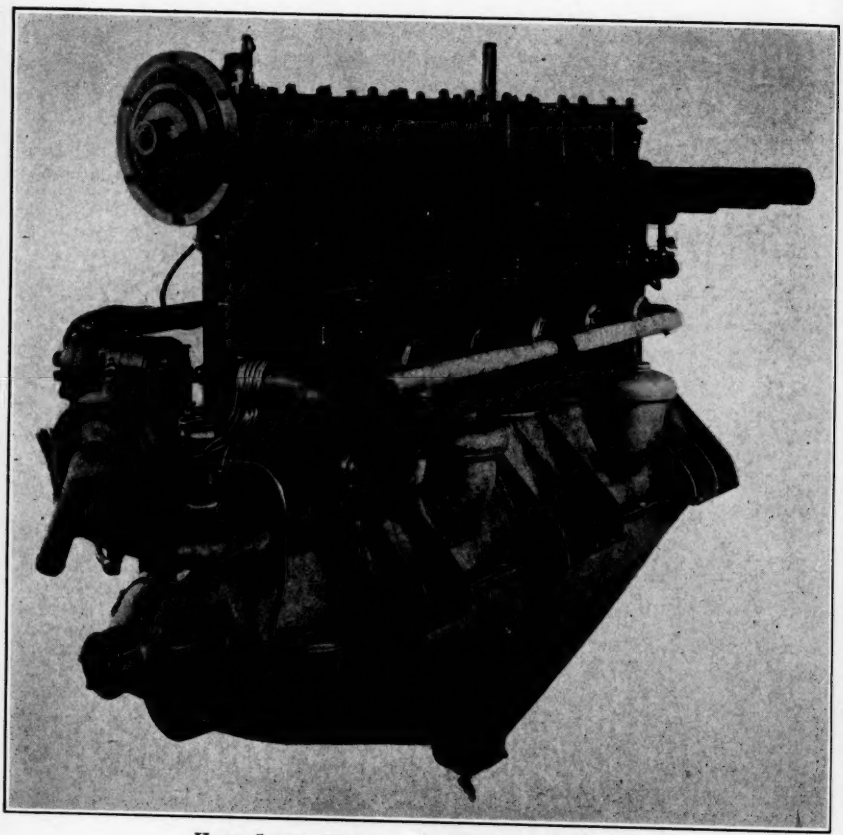
Cylinders.—Six cylinders are cast separately from a special mixture of grey and Swedish iron, having cylinder head with valve seats integral. Special attention has been given to the design of the water jacket around the valves and head, there being two inches of water space above same.

The cylinder is annealed, rough-machined, then the inner cylinder wall and valve seats hardened and ground to mirror finish. This adds to the durability of the cylinder and diminishes a great deal the excess friction.

Great care is taken in the casting and machining of these cylinders, to have the bore and walls concentric with each other. Small ribs are cast between outer and inner walls to assist cooling as well as to transfer stresses direct from the explosion to hold-down bolts which run from steel main-bearing caps to top of cylinders.



ROBERTS, MODEL 6X, AEROPLANE MOTOR.



HALL-SCOTT, MODEL A-5, AEROPLANE MOTOR.

The cylinders are machined upon the sides so that when assembled on the crank case with grooved hold-down washers tightened they form a solid block, greatly assisting the rigidity of crank case.

All cylinders are finished with dark blue baked enamel.

Connecting rod.—The connecting rods are very light, being of the I-beam type, milled from a solid chrome-nickel hand forging. The caps are held on by two ½ inch, 20-thread chrome-nickel through bolts.

The rods are first roughed out, then annealed. Holes are drilled, after which the rods are hardened and holes ground parallel with each other. The piston end is fitted with a gun-metal bushing, while the crank-pin end carries two bronze serrated shells, which are tinned and babbitted hot, being broached to harden the babbit. Between the cap and rod proper are placed laminated shims for adjustment.

Crank Case and Oil Sump.—Crank cases are cast of the best aluminum alloy, hand scraped and sand blasted inside and out. The lower oil case can be removed without breaking any connections, so that the connecting rods and other working parts can readily be inspected. This lower case is capable of holding seven gallons of oil. An extremely large strainer and dirt trap is located in the center and lowest point of the case, which is easily removed from the outside without disturbing the oil pump or any working parts.

Carburetion.—A double Zenith carburetor, having one float chamber, is provided. Automatic valves and springs are absent, making the adjustment simple and efficient. This carburetor is not affected by altitude. It is bolted directly to the engine base from which it receives its warm air. This also tends to keep the crank case cool. A Hall-Scott device, covered by U. S. Patent No. 632,919, allows the oil to be taken direct from the crank case and run around the carburetor manifold, which assists carburetion as well as reduces crank-case heat.

Starting Crank.—The starting crank is mounted in a com-

pact aluminum housing securely bolted to main crank case, thus forming an integral part of the motor.

Gears.—All gears, with the exception of the two bronze oil-pump gears, are of chrome-nickel steel. They are turnel up from extremely light forgings, with planed teeth, and are then hardened, the bore and faces being ground true. All gears where possible are bolted to flanges or made integral with shaft. All gears are enclosed and run in oil.

Magnetos.—Two waterproof six-cylinder Bosch magnetos are provided. One of these, being equipped with the dual system, affords easy starting. Both magneto interruptors are connected to a rock shaft integral with the motor, making outside connections unnecessary.

It is worthy of note that with this independent double-magneto system one complete magneto can become indisposed and still the motor will run and continue to give good power.

Piston Assembly.—The pistons are cast from the same semi-steel mixture as the cylinders. These are extremely light, yet provided with six deep ribs under the arch head, greatly aiding the cooling of the piston as well as strengthening it. The piston-pin bosses are located very low in order to keep the heat from the piston head away from the upper end of the connecting rod, as well as to arrange them at the point where the piston fits the cylinder best.

These pistons are fitted with three rings ¼ inch in width. A large-diameter heat-treated chrome-nickel wrist pin is provided, while two hardened tapered screws are assembled in such a way as to assist the circular rib between the wrist-pin bosses to keep the piston from being distorted from the explosions.

Pistons are first annealed, machined close to size, then hardened and ground to a mirror finish. These are all lapped into the cylinders after the rings have been installed.

Oiling System.—The oiling system is known as the high-pressure type, oil being forced to the under side of the main bearings under a pressure of from 5 to 30 pounds. This

system is not affected by extreme angles obtained in flying, or whether the motor is used for push or pull machines.

A large gear pump is located in the lowest point of the oil sump. Being submerged at all times with oil does away with troublesome stuffing boxes and check valves.

The oil is first drawn from the strainer in oil sump to the long jacket around the intake manifold, then it is forced to the main distributor pipe in crank case which leads to all main bearings. A by-pass located at one end of the distributor pipe can be regulated to provide any pressure required, the surplus oil being returned to the case.

Independent of this system, a small six-plunger pump feeds oil to each individual cylinder, being regulated entirely by the stroke of the plungers, which can be separately set.

A special feature of this system is the dirt, water and sediment trap, located at the bottom of the oil sump. This can be removed without disturbing or dismantling the oil pump or any oil pipes.

The lower crank case can also be removed without breaking any connections, as all joints are made through flange of case.

A small oil-pressure gage is provided, which can be run to the aviator's instrument board. This registers the oil pressure and also determines its circulation.

Cooling System.—The cooling of this motor is accomplished by the oil as well as by the water, this being covered by U. S. patent No. 632,919. This is accomplished by circulating the oil around a long intake manifold jacket. The carburetion of gasoline cools this regardless of weather conditions. Crank-case heat is therefore kept at a minimum regardless of weather conditions.

The uniform temperature of the cylinders is maintained by a very ingenious internal-outlet pipe, running through all six cylinders, having no connections, the joints being made with packing nuts. Slots are cut in this pipe, necessitating cold water being drawn directly around the exhaust valves. Extra large water jackets are provided upon the cylinders. Two

inches of water space is left above the valves and cylinder head.

The water is circulated by a large centrifugal pump, insuring ample circulation at all speeds.

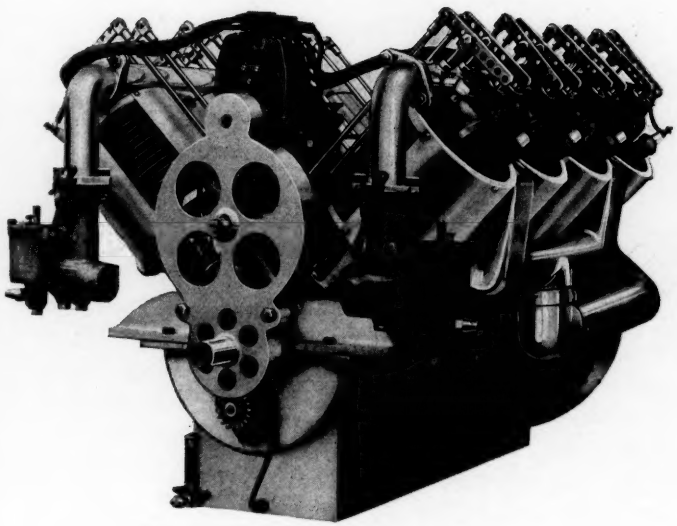
Crank Shaft.—The crank shaft is of the seven-bearing type, being cut from a billet of the highest-grade chrome-nickel-steel having a tensile strength of 275,000 pounds. The billet is first drilled, then roughed out, with crank pins on one side. The shaft is then heated and offset so that the cranks are 120 per cent. apart. After this the shaft is straightened, turned down to a grinding size, heat-treated, and then finished accurately to size.

The bearing surfaces are of extremely large size, this being accomplished without overdue length to the entire motor by offsetting the webs of the shaft. The offset acts as an oil scooper for oiling the connecting rods. This feature is found in no other motor. The crank-shaft bearings are 2 inches in diameter by 2½ inches long, excepting the rear main bearing which is 5 inches long.

Two thrust bearings are installed on the propeller end of the shaft, one for pull and the other for push. The propeller is driven by the crank-shaft flange, which is securely held in place upon the shaft by six keys. These drive an outside propeller flange, the propeller being clamped between them by six through-bolts. The flange is fitted to a long taper on the crank shaft. This enables the propeller to be removed without disturbing the bolts.

Timing gears and starting ratchets are bolted to a flange turned integral with shaft.

Cam Shaft.—The cam shaft is of the one-piece type, cams, air-pump eccentric and gear flange being integral. It is made from a low-carbon chrome-nickel forging. It is first roughed out and drilled its entire length, then cams are formed, after which it is case-hardened and ground to size. The four cam-shaft bearings are extra long and are made from Parson's



KEMP, MODEL J-8, AEROPLANE MOTOR.

white brass. A small clutch is milled in gear end of the shaft to drive any type indicator.

The cam shaft is enclosed in an aluminum housing bolted directly on top of all six cylinders, being driven by a vertical shaft in connection with bevel gears. This shaft, in conjunction with rocker arms, rollers and other working parts, are oiled by forcing the oil into the end of the shaft, using same as a distributor, allowing the surplus supply to flow back into the crank case through hollow vertical tube. This supply oils the magneto and pump gears.

Valves.—Extremely large tungsten valves, one-half the cylinder diameter, are seated in the cylinder heads.

Large-diameter oil-tempered springs held in tool-steel cups locked with a key are provided. The ports are very large and short, being designed to allow the gases to enter and exhaust with the least possible resistance.

These valves are operated by overhead one-piece cam shaft in connection with short chrome-nickel rocker arms. These arms have hardened tool-steel rollers on cam end, with hardened tool-steel adjusting screws opposite. This construction allows accurate valve timing at all speeds with least possible weight.

KEMP MODEL J-8 MOTOR.

This is an 8-cylinder, "J" type, four-cycle motor; air cooled; rated at 80 horsepower; bore 4¼ inches; stroke 4¾ inches; normal r.p.m. at full power, 1,150; weight, 380 pounds.

Oil consumption, 7 pints per hour.

Fuel consumption, 4.4 gallons per hour.

Cooling is accomplished by means of a fan mounted directly on the crank shaft, which discharges the air into two manifolds leading to the cylinder jackets. The distance the air travels to each cylinder is equal. On the cylinders are accurately machined radiating fins, around which the air from the fan passes.

Cylinders are semi-steel castings of the valve-in-head type, machined all over except the tops of the cylinders outside.

Valves are of tungsten steel.

The crank case is made from a very strong but light aluminum alloy, well ribbed. It supports the five main crank-shaft bearings which are lined with hard, high-speed, nickel-motor babbitt.

Two large ball-thrust bearings take the thrust load in either direction.

The crank shaft is machined from a solid billet of vanadium-steel, which is first blocked out, rough turned, heat treated, and finished without forging. This insures an absolutely flawless shaft, with all forging strains eliminated entirely, and the heat treatment insures great strength and sufficient hardness to prevent wear.

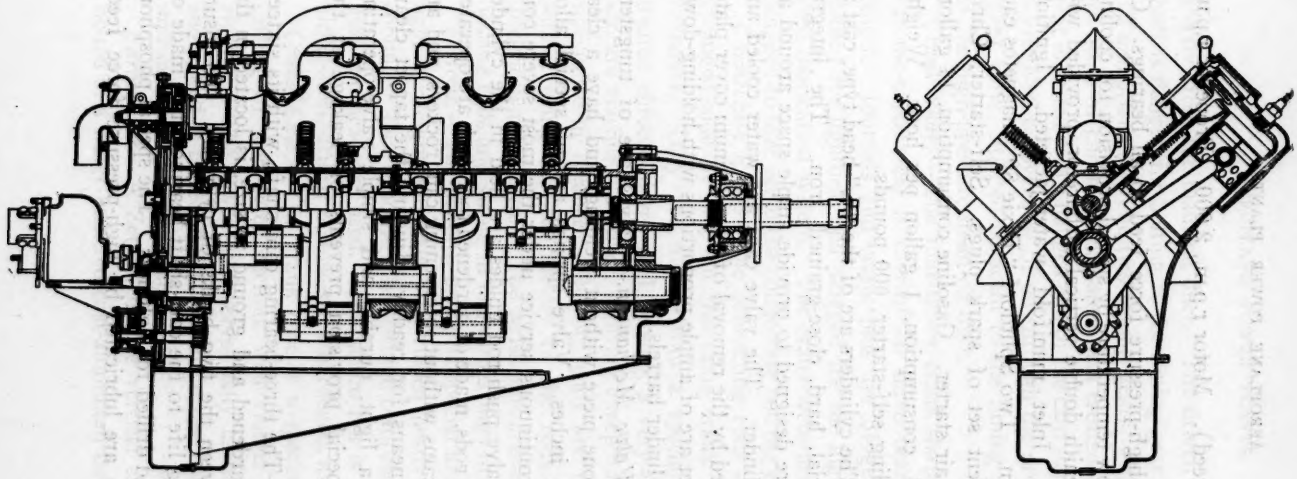
The cam shaft and cams are made from a very high-grade steel for this use, and runs in five bronze bearings, entirely enclosed within the crank case.

The oil sump holds four gallons of lubricating oil which is continuously pumped up and distributed over the cam-shaft bearings and main crank-shaft bearings, and in turn is splashed in the cylinders uniformly and then overflows back to the oil sump. This insures positive lubrication to all bearings and cylinders uniformly, which is one of the most important factors in the efficiency and reliability of aeronautic motors.

The pistons are made from the same semi-steel stock as the cylinders. They are very light, well ribbed and contain two Wasson piston rings. These piston rings are turned and ground uniform in thickness, and the tension is obtained by a special peining process which makes the tension uniform on all sides, the tension remaining uniform regardless of the wear on the outer surface of the ring.

THOMAS AEROMOTOR.

Type, 8-cylinder "V" four-cycle, water cooled. Propellers driven through reduction gearing. Bore, 4 inches. Stroke, 5½ inches. Brake horsepower at propeller shaft, 135 at 1,200



MODEL 8 135 H.P.

THOMAS AEROMOTOR CO.

ITHACA, N.Y.

r.p.m. (guaranteed). Motor r.p.m., 2,000. Propeller r.p.m., 1,200.

Lubrication, high-pressure force feed to all bearings. Oil maintained at low temperature by special provision for cooling. Carburetion, Zenith double vertical carburetor, provided with hot-air intake. Inlet manifold water-jacketed. Ignition, Duplicate system. Two Splitdorf "Dixie 80" magnetos each firing independent set of spark plugs. Self-starter, Christensen gasoline air starter. Gasoline consumption, 14 gallons per hour. Oil consumption, 1 gallon per hour. Weight, complete, including self-starter, 600 pounds.

Cylinders.—The cylinders are of the "L" head type, cast in pairs of a special, hard, close-grained iron. The integral water jackets are designed to provide ample space around all parts of the cylinder. The valve caps are water cooled and are easily reached by the removal of an aluminum cover plate. The cylinder feet are of ample proportions with holding-down bolts close to cylinder barrels.

Valves and Valve Mechanism.—Valves are of tungsten-steel, made in one piece without welding, and have a clear opening of 2⅛ inches. Valve springs are of special alloy steel to insure continuous service under the most severe conditions. The valve push-rod guides are held in the cylinder feet. The push rods, made of hardened tool steel, are operated directly by the cams without intervening cam rockers, and are provided with means for readily adjusting the tappet clearances, making a light, durable and simple valve-operating mechanism. Special provision prevents oil leakage at the push rods.

Cam Shaft.—The three-bearing cam shaft, with its sixteen integral cams hardened and ground to size, is located in the crank case between the two banks of cylinders. To insure rigidity and long life to the cam-shaft bearings it is made of generous size and drilled for lightness. The special phosphor-bronze bearings are lubricated by high-pressure force feed,

directly from the main oil duct. The driven end of the cam shaft is flanged for bolting to the timing gear.

Timing Gears.—The two timing gears, with spur-cut teeth. are made of chrome-nickel-steel heat treated. The cam-shaft gear is bolted to the flanged end of the cam shaft. Lubrication is provided by overflow of oil through the pressure-regulating relief valve, situated adjacent to the timing gears at the end of the high-pressure main oil duct.

Crank Shaft.—The four-throw crank shaft made of special chrome-nickel-steel is carried on three bearings of liberal length. Journals and crank pins are of generous diameter, drilled for lightness and plugged for the high-pressure force-feed lubrication system. The crank webs are also drilled and plugged to lead oil to the connecting rod big ends.

Connecting Rods.—The connecting rods are of H section, arranged side by side on the same crank pin for opposite cylinders. They are made from a special grade of heat-treated chrome-nickel steel, having, a very high elastic limit, and are machined all over, securing exceptionally light weight and uniform balance. The big ends are babbitt lined, the babbitt being applied directly to the rods and forming an integral part of them. The wrist pins are locked in the connecting rods and swing in liberal bearings in the piston bosses.

Pistons.—The pistons, made of special aluminum alloy, are light in weight and have well ribbed heads for strength and cooling. Two concentric lap-jointed compression rings located near the head are used. The piston pin is made of chrome-nickel-steel, drilled hollow, pack hardened, and ground to size.

Crank Case.—The crank case is a special alloy aluminum casting of deep-sided section, well ribbed for strength and rigidity. The lower half serves only as an oil reservoir or sump, and is bolted directly to the upper half. It has a four and one-half gallon capacity.

Carburetor.—A Zenith, double vertical carburetor is used, bolted to a water-jacketed cast-aluminum double-branch mani-

fold serving the two banks of cylinders. All manifold bends are of large radii. Hot air intake is fitted to the carburetor.

Lubricating System.—The lubricating system is of the high-pressure, circulating type. The main supply is carried in the aluminum sump, covered by a large fine-mesh wire screen which insures clean oil at all times. A gear pump, located within the crank case and driven from the crank-shaft timing gear, takes oil from the sump and delivers it to a main oil duct, also within the crank case. Drilled holes in the main bearing-supporting webs, connecting with this duct, lead the oil directly to the three-crank shaft and cam-shaft journals. The connecting-rod big ends receive oil through the drilled crank-shaft webs and crank pins. The wrist pins and pistons, including cams and push rods, are lubricated by the oil thrown off the crank pins.

Oil-Cooling System.—A comprehensive oil-cooling system, entirely independent of the lubricating system, is also provided. Oil is taken from the sump, above the two-gallon capacity level, by a separate gear pump and passed through cooling coils and delivered to the sump again. To serve as a telltale, this oil may be passed through a sight gage, under the eye of the pilot, and so warn him when the two-gallon level has been reached. This telltale will also prove valuable in the case of flights of long duration, where an oil-supply tank is necessary, by warning him when to add oil to the sump.

Water-Cooling System.—A single centrifugal water pump is operated at crank-shaft speed by a shaft located above the cam shaft and driven by a steel gear meshing with the cam-shaft timing gear. Equal water distribution to both banks of cylinders is insured by a siamese outlet from the pump to the brass water pipes at the base of the cylinder jackets. Water-outlet connections are arranged over each exhaust valve. All water pipes are fitted close to the cylinders for rigidity and economy of space. Only four hose connections are necessary with each motor.

Ignition System.—Two Splitdorf "Dixie 80" magnetos are

ASHMUSEN, 105-H.P., AEROPLANE MOTOR.

used, each constituting an entirely independent ignition system, with two spark plugs per cylinder. They are arranged side by side, and are driven at crank-shaft speed by spur gearing at the rear end of the crank shaft. This position insures their operation under the most favorable conditions, besides providing ready accessibility.

Propeller Drive.—The propeller shaft is driven from the crank shaft through two spur gears of special chrome-nickel gear steel, heat treated. It is rigidly supported by two large ball bearings housed in a well ribbed aluminum gear case bolted to the crank case. Both ball bearings are of the annular type, the one near the propeller end of the shaft having a two-row ball race for taking the combined radial load and thrust from the propeller in both directions. The propeller shaft is drilled to secure light weight.

Self-Starter.—A Christensen combined gasoline and air-starting system is located between the two banks of cylinders and is driven at crank-shaft speed by the same shaft that operates the water pump. In its application to the Thomas motor no drive shaft is exposed, thus eliminating all chances of oil leakage.

The starter is composed of four principal parts—an air pump in operation only when compressing air for storage purposes; an automatic distributor which is in action during the starting period only; an automatic carburetor which supplies gasoline to the air distributed to the cylinders; and a compressed-air tank. The entire apparatus weighs approximately 40 pounds.

ASHMUSEN MOTOR.

This motor is of the horizontal, opposed-cylinder type; air cooled; 12 cylinders; bore, 3.75 inches; stroke, 4.5 inches; horsepower, 105 at 1,800 r.p.m.; weight about 380 pounds.

The motor is equipped with two Franklin carburetors and full double manifolds. The air is drawn through flutes on the side of the cylinders to the manifolds, thence to the

carburetors. This system accomplishes two purposes: it cools the cylinders and warms the air to the carburetor. Delco ignition is used. The propeller is driven from the cam shaft at one-half motor speed. The fly-wheel weighs 14 pounds. The crank case is of aluminum alloy. The crank shaft is forged from nickel-steel and is heat treated. The cylinders, cylinder heads and pistons are of grey iron. The cylinders weigh 6 pounds each and the pistons 2 pounds. The valves are of tungsten steel. Main and cam-shaft bearings are of the ball type. Wrist-pin bearings are of Parsons white brass.

A compression release is fitted to hold the exhaust valves off their seats. The arrangment of the cylinders permits a perfect balance of moving parts and reduces the vibration to a minimum.

One of the greatest features of this motor is its accessibility; it is possible to remove all cylinders and replace them in 45 minutes.

GYRO "DUPLEX" MOTOR.

This is a 9-cylinder, air-cooled rotary motor, developing 100 H.P. (normally) at 1,130 r.p.m.; bore, 4½ inches; stroke, 6 inches; weight, 270 pounds.

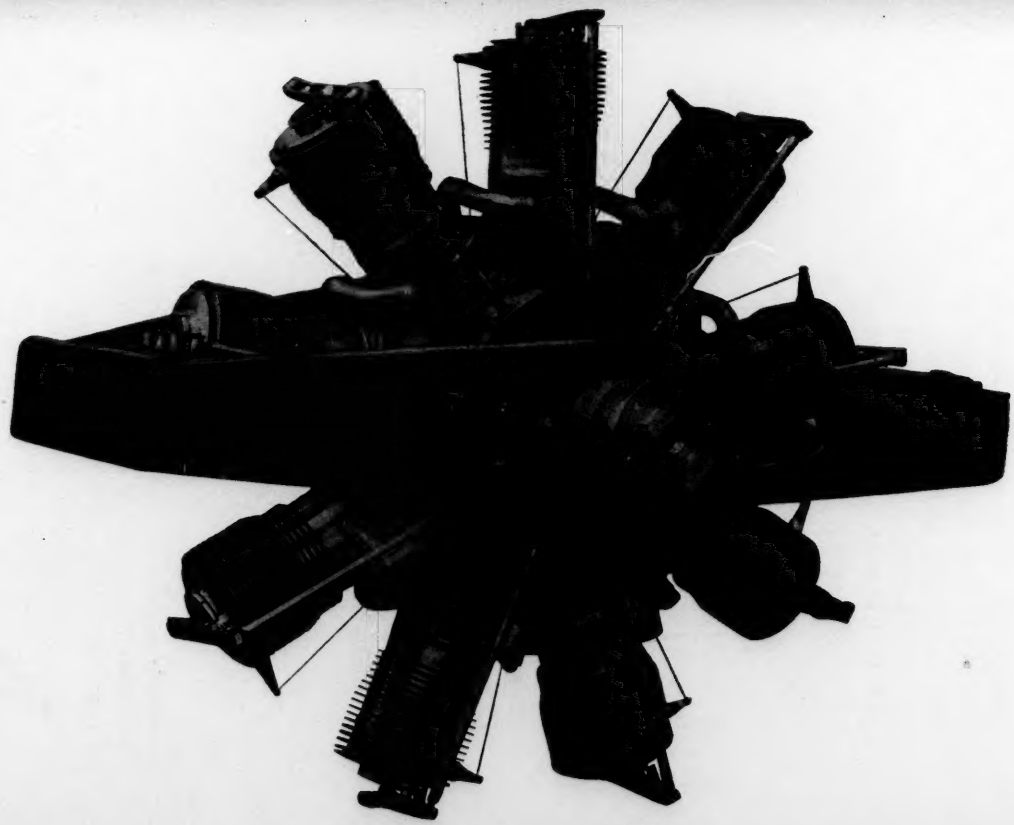
Specifications.

Cylinders are turned from solid billets of special alloy steel and ground to size. Bore, 4½ inches, stroke, 6 inches. All cylinders are carefully balanced to eliminate vibration.

Cooling is by direct radiation, which is increased by turning integral flanges on the cylinder walls. This system has proven adequate under the most adverse conditions and has several advantages over the indirect, or water cooling, systems, not the least of which is a clear saving of 100 pounds in weight.

Pistons are of the best-grade cast iron, fitted with oil deflectors for economy, and two rings of patented GYRO design.

Connecting Rods are of vanadium-steel, heat treated and



GYRO "DUPLEX" AEROPLANE MOTOR.

machined all over. Connecting rod assembly or "spider" is carefully balanced.

Crank Shaft is of nickel-steel, drilled out for lightness and to serve as a channel for feeding gas and oil to the crank case.

Crank Case is of vanadium-steel, made in two halves, permitting the cylinders to be clamped between the two halves.

Bearings are of the ball, roller, or plain type in various parts of the motor. Each is selected as being best adapted to the work, regardless of cost.

Cam is "Duplex" type, made of the finest die-steel. It is driven through a differential gear train.

Ignition is by Bosch high-tension magneto direct-to-spark plugs, via a very simple distributor of special design.

Carburetion is semi-automatic, gasoline being fed to crank case through a gear pump and spray nozzle.

Oiling System is automatic and absolutely positive, being driven to the central bearing by a gear-driven displacement pump. From there it passes through ducts provided for the purpose, to all points of friction.

Gyro motors are equipped with a variable compression cam which allows the motor to be started readily and to "idle" at 250 to 300 r.p.m. They are further distinguished by the following features: No back pressure nor back firing; no intake valves to grind; no oil waste through piston head.

Neither pistons nor cylinders have any reciprocating motion, but simply revolve each about a separate center.

Description and Operation.

The inside of the motor is bare of all accessories save the pistons P, the connecting rods R, and the crank shaft. In this respect it resembles the ordinary reciprocating motor.

The main exhaust valve D remains on the top of the cylinder and is operated by rod and cam. This cam is now in "duplex" form, one side operating the main exhaust, the other the slide intake mechanism B, C; the latter is attached to the outside of

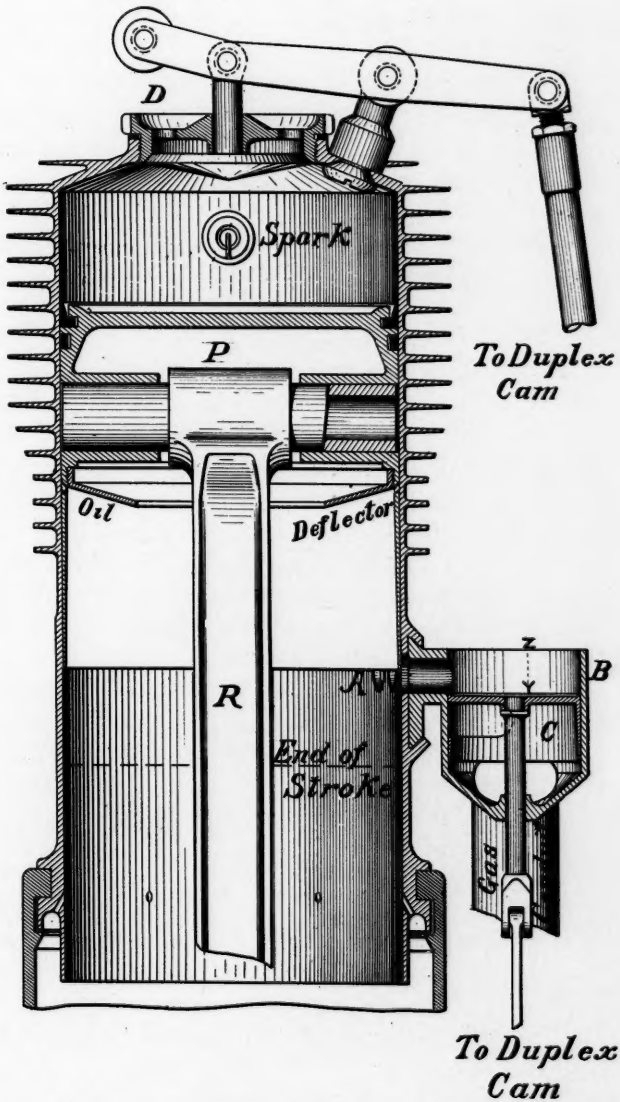
the cylinder about two inches above the end of the power stroke and is readily detachable.

At this point there are provided the usual Gyro auxiliary ports A, through which the main pressure of the nearly-spent stroke exhausts itself, thereby reducing the pressure necessary for opening the main exhaust valve.

Outside of these ports is a cage, B, in which a small hollow slide, C, moves with a stroke of about half inch, y to z; this stroke depends upon the shape of another cam forming a twin to the main exhaust cam.

The operation is as follows: When the power stroke reaches the auxiliary ports A the gases escape and relieve the pressure in the cylinder. The piston continues 1½ inches, to the end of the stroke, and then returns for scavenging the burnt gases out through the main exhaust D; the piston then moves down for the intake.

The exhaust D remains open until just before the piston on its intake stroke reaches the auxiliary ports A. In the meantime the small intake slide C has moved outward to z and the auxiliary port A is now connected through the cage B to a gas conduit filled with fresh mixture. The main exhaust has closed and the piston moves 1½ inches farther and sucks the gases into the cylinder. The intake slide C then returns to its original position, while the piston moves outwards in the compression stroke.



SECTION THROUGH CYLINDER—GYRO "DUPLEX" AEROPLANE MOTOR.

TESTING OF STEAM TRAPS AT THE U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

By Lieutenant (Junior Grade) J. L. Kauffman, U. S. N., Member.

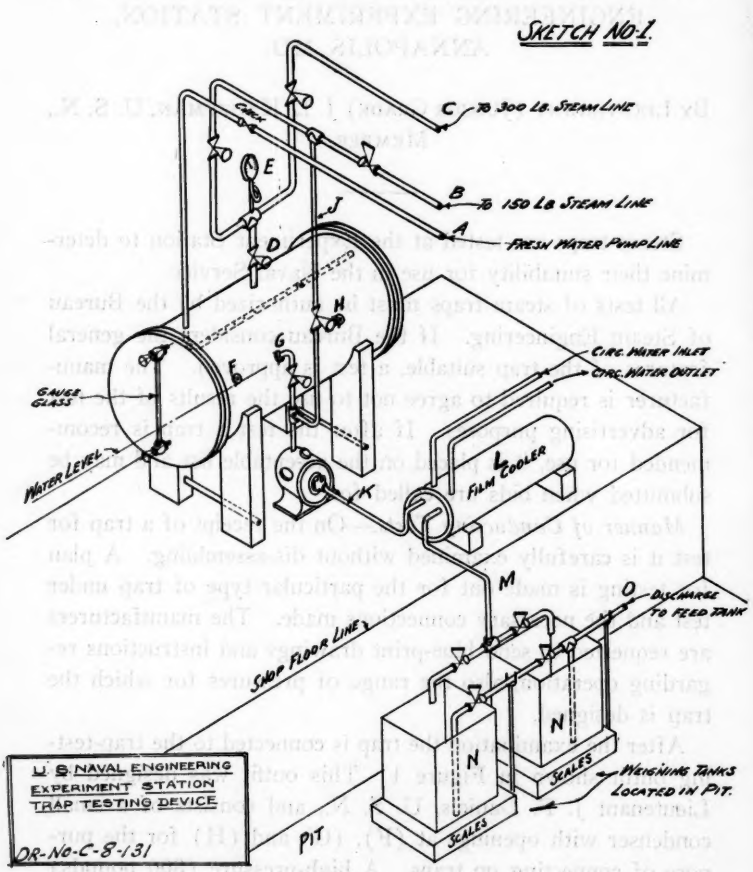
Steam traps are tested at the Experiment Station to determine their suitability for use in the Naval Service.

All tests of steam traps must be authorized by the Bureau of Steam Engineering. If the Bureau considers the general features of the trap suitable, a test is approved. The manufacturer is required to agree not to use the results of the test for advertising purposes. If after the test a trap is recommended for use, it is placed on the acceptable list and may be submitted when bids are called for.

Manner of Conducting Tests.—On the receipt of a trap for test it is carefully examined without dis-assembling. A plan for testing is made out for the particular type of trap under test and the necessary connections made. The manufacturers are requested to send blue-print drawings and instructions regarding operation, also the range of pressures for which the trap is designed.

After the examination the trap is connected to the trap-testing outfit shown in Figure 1. This outfit was designed by Lieutenant J. F. Daniels, U. S. N., and consists of a small condenser with openings at (F), (G) and (H) for the purpose of connecting up traps. A high-pressure (300 pounds) steam line and a low-pressure (150 pounds) steam line are connected for use with the different traps. A gage (E) indicates the condenser pressure. The discharge from the trap passes through the film cooler (L) and thence to the weighing

tanks. The cooler is for use on capacity tests when it is necessary to obtain accurately the amount of water passing through the trap. During functioning tests the discharge line is broken at the film cooler, to better observe whether or not steam or



water dribbles through. Connections are made on (K) and (D), which lead to the recording gages, in order to give a continuous record during capacity and endurance runs.

After connecting up the trap the following tests are made:

Functioning Test.—This test consists of runs made at pressures ranging from 25 to 300 pounds gage, to determine the ability of the trap to function through the range of these pressures. To make this test the condenser is first half filled with water, and the steam turned on. The valve mechanism is then held open, allowing the water and a small amount of steam to be blown through, in order to clear the passages and heat up the trap. The condenser is again filled with water to the height of the trap connection (marked water level in Figure 1), steam turned on and runs made at 25, 50, 100, 150, 200, 250 and 300 pounds per square inch gage. On each run observations are made as to whether or not the trap takes care of the condensate; the amount of steam blown through, if any; the tightness of the valve mechanism, and the ability of the trap to function regularly over a range of pressure without changing the adjustment.

Capacity Test.—This test consists of four or five runs at the pressures given above, to determine the amount of water, in pounds per hour, that the trap will discharge at each pressure when working continuously. The test is made with the condenser about half filled with water, so that the trap will work at its maximum capacity. Some traps are also tested by connecting the pump discharge direct to the trap, in order to get the maximum amount of water possible for the trap to discharge.

Limited Endurance Test.—This consists of runs of eight hours or more over the usual range of pressures, to find out if the trap will maintain its ability to function throughout that time. Recording gages are placed on the pressure and discharge sides of the trap to give a continuous record of its performance. Careful watch is kept in order to make sure that the trap handles all of the condensate and that no water or steam leaks past the valve.

Endurance Test.—If the previous tests indicate the trap to be satisfactory, the trap is installed in the power plant of the Experiment Station for at least three months to determine

whether or not the trap will function over a relatively long period with a minimum amount of care and overhauling. A record is kept of every time that the valve sticks and of the number of times necessary to overhaul the trap.

Special Tests.—Some types of traps require special tests. Ball-float traps are tested to determine their ability to function properly when a ship is rolling. To make this test the trap is mounted on rockers and connected to the testing outfit by flexible steam hose. The trap is then oscillated about 35 degrees on a side, and observations are made of its ability to function under these conditions. The ball float is also separately tested by putting the ball in a cylinder, and connecting the cylinder to a hydraulic test pump. Pressure is slowly applied until a pressure of 550 pounds gage is reached. A careful examination is made of the ball to discover if the ball leaks at this high pressure.

The traps tested at the Experiment Station are divided into four classes, namely:

(*a*) Bucket traps;
(*b*) Ball-float traps;
(*c*) Differential traps;
(*d*) Expansion traps.

Bucket traps are in general use throughout the naval service, and their principal features are familiar to most officers. The various makes differ chiefly in the valve mechanism. With few exceptions all have a main and an auxiliary valve, but the methods by which these valves are controlled are widely different in the various types.

Ball-float traps have not been in general use in the naval service, due to the inability of some types to function when oscillated and to the failure of the ball floats when subjected to high pressures. Due to the uniform rise and fall of the float in traps of this type, only one valve with positive action is necessary. As with bucket traps, the various designs differ principally in the valve mechanisms.

Expansion traps have been used in the service at various times and are widely used on heating systems. There are a large variety of expansion traps on the market, but most of them are cheap and unreliable. The best known type depends on the expansion and contraction of a brass pipe, with change of temperature, to open and close the valve. Some use both iron and brass pipes, the difference in expansion of these two metals actuating the valve. Another, and seemingly very popular type for heating systems, consists of a hermetically sealed tube filled with an expansive liquid, the action of this liquid at the various temperatures moving the valve. A similar type is one using a hollow crescent-shaped tube very much like the Bourdon type in pressure gages.

Differential traps are not so well known as the others, and have never been used in the service. In this type the valve is opened and closed by the variation in pressure on a diaphragm due to the difference in the height of water in the inlet tubes.

In summing up the results of tests of all types of traps the following points are considered desirable:

Substantial construction of all parts.

Comparatively simple operation.

Accessible working parts.

Positive action of the valve mechanism.

Sufficient capacity.

Functioning regularly when working to capacity.

Proper functioning over a range of pressure.

Efficient steam and water joints; easy to keep tight.

Gage glass installed on trap to see at a glance if trap is working properly.

Reliable hand mechanism for use in case valves stick.

Location of valves so that no mud or scale can settle on them.

Durability and ability to function regularly in service for long periods without overhauling.

Water seal for valve mechanism.

Strainer at inlet.

The following disadvantages are frequently found:

Constant dribbling of steam.

Valve chamber at bottom of trap, which acts as catch-all for sediment.

Knife-edge valve seats, making it very difficult to keep valves tight.

Small and winding water passages, where main and auxiliary valves are used.

Large bulk and weight with relatively small capacity.

Bucket not buoyant enough to overcome friction of parts.

Main valve not properly balanced.

Ball-float leaks or will not stand high pressures.

Trap will not operate when oscillated.

Change in adjustment necessary for change in room temperature on expansion traps.

In the design of traps it would be very desirable to have a valve in which the erosion does not occur on the faces of the valve and seat as might be arranged with a cylindrical valve. This has been tried, but the faulty design of the other parts caused the rejection of the trap.

Although a great many excellent traps are found on the market, it is very difficult to obtain a trap that will operate satisfactorily when placed in the bilges of a ship or where it is not easy to inspect, especially when a trap must function regularly for long periods without receiving much attention. On those lines where trapped water continually accumulates, in lieu of traps it is considered better practice to install a gage glass and drain valve, to be operated by the man on watch.

CLASSIFICATION OF SCRAP MATERIAL AT THE NAVAL GUN FACTORY.

By E. W. Bonnaffon, Pay Inspector, U. S. Navy.

The following outline is given of the extent of the classification of scrap material at the Naval Gun Factory. This phase of industrial economy has primarily to do with the segregation or sorting of this material, either for further use or future sale.

The efforts made by the gun factory along these lines have simply kept pace with similar activities in other commercial establishments.

The underlying purpose is two fold: to so sort the scrap as to permit its being used advantageously in the plant (that is as raw material in manufacture), or so that it may be sold at a higher price.

The question always uppermost in deciding just how far to go in this work must be "Will it pay?" To answer this intelligently the cost of handling, sorting and storing must be considered together with the possibilities of its use, and the probability of the proceeds of sale if disposed of in that way. This requires a knowledge of the market prices of these various materials and also a knowledge of industrial operations.

The advice and suggestions made by the Inspection Department have been most valuable and this coöperation has materially assisted the systematic handling and use of scrap.

The primary "don't" in this business is not to mix scrap. This lies as the foundation of all later results. To illustrate the importance of this first step it must be borne in mind that all turnings from the shops, which are mixed, as, for instance, brass and bronze turnings with steel, are of relatively little value. Each kind of metal must be kept absolutely separate.

It is possible to divide the metals by means of a magnetic separator into ferrous and non-ferrous, and this is done whenever necessary, but it is not feasible to further divide the fine particles of brass and bronze, etc. They should be kept from steel turnings, however, as their presence reduces materially the value. Hence the necessity of placing the trays right at the machine to receive the turnings as they come from the work.

Any mistakes made through ignorance or carelessness are costly. Valuable scrap material is often mixed so that it is either unfit for use or cannot be sold to advantage. To accomplish this satisfactorily the separation must be made as near the work as possible. For instance, suitable receptacles are distributed through the shops and, when possible, at the machines, to receive the different kinds of scrap, since it is just as easy to place the material separately at the start of the process, and then the character of the material is absolutely known, rather than later, when it would be necessary to add additional labor to sort it out, and the kind of material might be in doubt.

To show that the accumulations of this material at the gun factory are sufficiently large and valuable enough to warrant special attention, the following list is given of the annual output, with the prices received from sale or its estimated value to the plant when used:

	Pounds.	Per lb.
Nickel-steel borings and turnings,	500,000	.005
Carbon-steel borings and turnings,	1,500,000	.003
Cast-iron borings,	100,000	.003
Carbon-steel (shoveling and charging stock),	500,000	.003
Cast-iron (usable sizes),	400,000	.005
Pipe and tubing,	100,000	.004
Burnt iron,	150,000	.005
High-speed steel,	1,700	.07
Miscellaneous large scrap steel and wrought iron,	1,000,000	.004

	Pounds.	Per lb.
Galvanized iron,	60,000	.0015
Sheet steel,	60,000	.0025
Cast iron (too large for use),	100,000	.003
Brass foundry ashes,	550,000	.003
Brass foundry skimmings,	50,000	.035
Broken crucibles,	50,000	.003
Dross,	101,000	.07
Shop sweepings,	90,000	.035
Emery grindings,	8,000	.10
Solder dross,	2,000	.05
Brass turnings,	200,000	.10
Ordinary bronze turnings,	200,000	.12
Manganese-bronze turnings,	40,000	.10
Cupro-nickel turnings,	10,000	.10
Copper,	20,000	.12
Ordinary bronze,	250,000	.12
Manganese-bronze,	50,000	.10
Brass,	100,000	.10
Waste paper, printed,	10,000	.0125
wrapping and sweepings,	20,000	.007
card board,	20,000	.007
Dirty and oily waste,	...	...
Copperas (ferrous sulphate),	...	...
Residue of oxy-acetylene plant,	5,000	.01
Rope and cuttings,	24,000	.025
Rubber,	4,500	.02
Leather,	2,500	.06

All of this material that can be utilized in the plants is turned into store at a price established by a board, and when drawn out is charged in the same way as new material. The credit given in this way to the shop or job is an incentive to the manufacturing department to save and to recover the waste.

The prices are based on a reasonable estimate of the market value of the actual contents plus the cost of putting into usable shape. For instance, brass turnings which analysis shows contains 70 per cent. copper and 30 per cent. zinc would de-

pend on the market price of copper and that of zinc, and in the ratio of 7 to 3, together with the cost of melting into ingots. The prices fixed ignore temporary fluctuations.

The nickel-steel turnings, practically all of which are obtained from work in the gun shop, upon being collected in large tubs are sent direct to the foundry for use in the open-hearth furnace.

The carbon-steel of suitable size (what would be termed in the trade as shoveling or charging stock) is also used in the open hearth.

The cast iron is placed in a separate bin and broken under a derrick installed for this purpose, and later used in the foundry.

All the rest of the steel and iron is sold.

The non-ferrous metals, which include turnings, scrap, etc., of brass, bronze, manganese-bronze, copper, etc., are collected and sorted in an enclosure adjacent to the foundry and near to the furnace in which it is to be melted. This material enters very largely into the manufacture of all brass and bronze castings, approximately 70 per cent. being used as against 30 per cent. new on purchased metals.

Baling machines are used to put the stringy or fluffy turnings into convenient shape for charging into the furnace, and a machine is being devised to compress the finer turnings and borings into briquettes, so that when it is of known analysis it may be melted directly in the crucible. To show the remarkably satisfactory results obtained in the use of scrap non-ferrous materials with—

800 pounds Tobin turnings,
1,000 pounds dross and skimmings,
800 pounds condenser tubes,
300 pounds manganese-bronze scrap,
500 pounds Tobin-bronze stampings and trimmings,
45 pounds aluminum-zinc scrap,
110 pounds cupro-manganese hardener,
260 pounds sheet-zinc scrap,
260 pounds slab zinc,

it has been possible to manufacture manganese-bronze of the following analysis and characteristics:

Chemical:

Copper, 59.88; zinc, 38.08; tin, .62; lead, .40; manganese, .08; aluminum, .66; iron, .28.

Physical:

Tensile strength.	Yield point.	Elongation.
68,805 pounds.	36,160 pounds.	26.1 per cent.
65,393 pounds.	39,126 pounds.	36.05 per cent.
67,990 pounds.	33,104 pounds.	30.05 per cent.

Waste paper is collected in the various offices and at the same time separated into classes for sale. Arrangements are being made to recover all oily or used cotton waste and clean and dry it for further use incidentally obtaining the oil for use.

Copperas is obtained as a by-product from the process of cleaning and treating (pickling) sheet metal with sulphuric acid. It will be sold, and is valuable as a disinfectant. Methods are also being devised for the recovery of bluestone or copper sulphate from the pickling bath in the cartridge-case shop.

The residue resulting from the manufacture of oxygen in the oxy-acetylene plant consists of muriate of potash and bin-oxide of manganese, which is a very valuable ingredient of fertilizers and is disposed of periodically for that purpose.

Rope, rubber and leather are obtained from various sources, principally from remnants resulting in manufacture.

MECHANICAL TEST STATION FOR A NAVY YARD.

By E. F. Maas, Associate.

In a letter dated June 10, 1914, the Bureau of Steam Engineering authorized an expenditure by the Navy Yard, Puget Sound, for installing and equipping a mechanical testing station. The work of designing the necessary apparatus and of purchasing such items as could most economically be obtained in that way was immediately proceeded with. At the present writing all, save one, of the main units of the station are completed, and some have been in operation for a considerable time.

The test station is located in one corner of the building which is reserved and now partly used for a machinery erecting shop. Its main units, as shown on Fig. 1, consist of the following: One 100-H.P. boiler with feed pumps and oil-burning apparatus; one steam-pressure reducing gear; one steam-drum for testing safety valves; one test gear for reducing valves; one steam-trap test set; two foundation plates for setting up machinery; one hydraulic brake; one condenser, with air pump and water-weighing tank; one switch board; one pump-testing set; one duplex hydraulic pump; two hydraulic-test tables; one test gear for chain hoists; one gage-testing set; one tensile-test machine; one oil-testing machine; water, air, steam, and oil piping connecting the various testing units; handling facilities, consisting of trolley system and column gib crane.

The boiler is of the Talbot manufacture, and rated at 100 H.P. It is designed for a working pressure of 400 pounds per square inch, and capable of delivering a considerable superheat even at this pressure. The boiler is oil-burning, and the oil-fuel apparatus so arranged that either the mechanical high-pres-

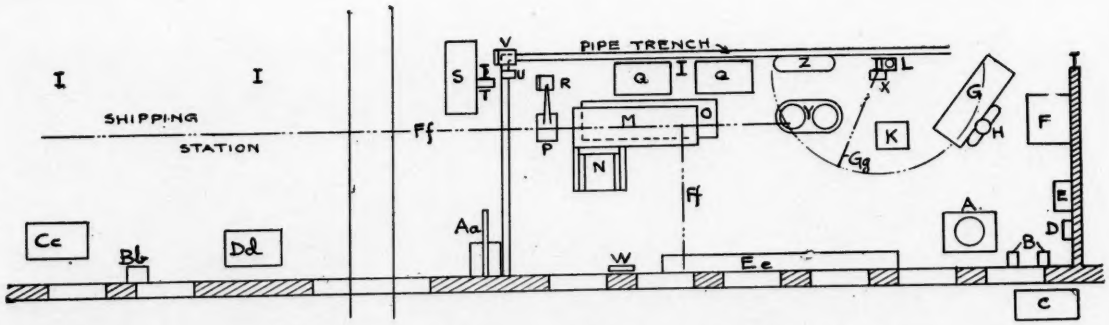
sure system or the steam system for atomizing the oil can be used. The apparatus includes: Two duplex horizontal Worthington pumps, 2 inches by 1⅛ inches by 2¾ inches, each one of which is capable of delivering all the oil consumed by the burner; one automatic governor and pressure regulator for pumps; one oil heater; one large air chamber in the discharge line, and one 420-gallon supply tank, with a small steam coil for heating the oil during cold weather. All of these parts will be seen on Fig. 2, with the exception of the supply tank, which is located outside the building, adjacent to the place where the oil-suction pipe is seen coming through the wall, and the regulator valve, which was installed after the photograph was taken.

Both the supply of oil to the burner and the amount of feed water entering the boiler are automatically controlled by throttle valves actuated by one of the boiler tubes acting as a thermostat. In addition to this a feed relief valve, which is operated by a pressure connection from the boiler, is provided. A peculiarity, and perhaps one of the difficulties, of this type of boiler is the great internal resistance it offers to the flow of water and steam, thereby making necessary a very high feed pressure. The greater part of this resistance is caused by the feed throttling just mentioned. When steam is delivered at 200 pounds pressure it is necessary to have the feed pump deliver at 400 to 500 pounds pressure, even though the feed pipe is very short, and no extra obstructions in the form of feed heater or grease extractor interposed.

The feed pumps are two simplex double-acting 6-inch by 3-inch by 8-inch steam pumps, each one capable of delivering the required feed to the boiler when it is steaming at full capacity. The steam ends for these pumps were obtained from two auxiliary feed pumps removed from the former U.S.S. *Davis*, and the water ends were manufactured by the navy yard. The water ends are cast-iron, bronze-bushed, and have steel ball valves. They have been tested to 1,500 pounds per square inch, and will deliver water up to 1,100 pounds

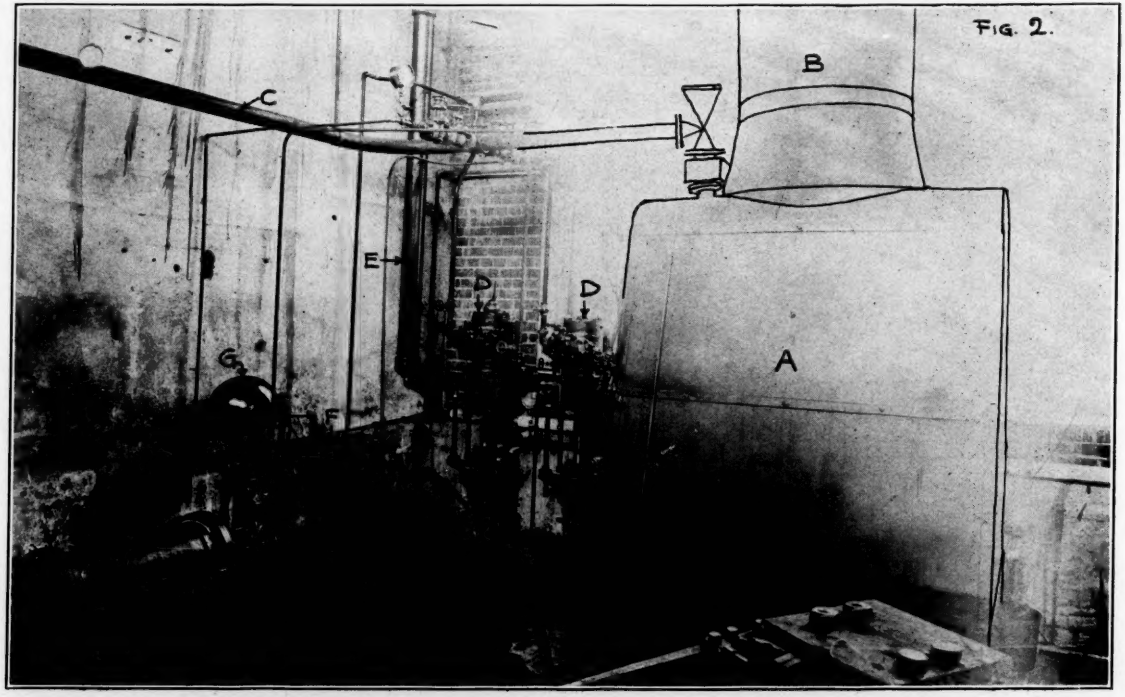
pressure with the steam pressure at 250 pounds. A large air chamber of 1,850 cubic inches capacity is provided for steadying the pressure of feed delivery. It can be connected at will to the discharge side of either pump. The steam-pressure-reducing gear consists of four 3-inch pressure-reducing valves with stop valves, relief valves and manifolds, arranged as shown in Fig. 3. This photograph had to be taken before the pipe covering was installed and will show the installation in a somewhat incomplete state. The reducing valves are designed to reduce from a pressure of 400 pounds per square inch to 300, 225, 150 and 80 pounds per square inch, respectively. Any steam pressure from the full boiler pressure down to 50 pounds per square inch, or less, can thus be obtained in the low-pressure manifold, by opening up the steam passage through the proper reducing valve and setting this valve to the pressure desired. For each reducing valve there is one 2-inch relief valve provided, on its low-pressure side, to insure that at no time will the steam pressure exceed that predetermined for the particular test. Gages and thermometers are provided as shown in the figure. From the low-pressure manifold connections are made to the various places in the testing room where steam is required.

Owing to the fact that the boiler has a very small steam space, it has been necessary to provide an extra steam drum for the purpose of giving the required volume of steam for testing safety valves. This drum with its connections and fittings are shown on Fig. 4. The dimensions of the drum are 2 feet 6 inches inside diameter by 8 feet long. On account of both its heads being concave, the useful cubic content is only 34 cubic feet, a capacity which has been found none too great for the larger-capacity safety valves. The steam connection from the reducing gear, shown on Fig. 3, is made to the bottom of the drum, as is also the drain connection. On top of the drum are shown pressure gage, thermometer, steam bleeder pipe connecting into the atmospheric exhaust, and two 3-inch quick-opening gate valves to which the valves to be

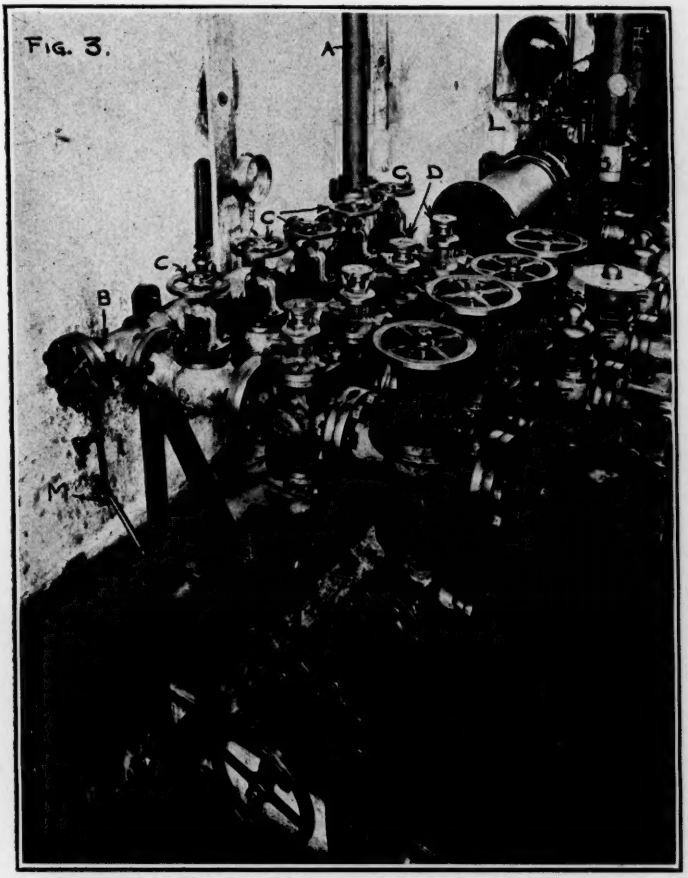


A = BOILER
B = FEED PUMPS
C = FUEL OIL TANK
D = OIL PUMPS
E = OIL HEATER
F = PRESSURE REDUCING GEAR
G = STEAM DRUM
H = ESCAPE PIPE
K = RED. VALVE TEST GEAR
L = STEAM TRAP " "
M = HOR. FOUNDATION PLATE
N = VERT. FOUNDATION PLATE
O = SUCTION TANK
P = HYDRAULIC BRAKE
Q = MEASURING TANKS
R = SCALE
S = CONDENSER
T = AIR PUMP
U = SURGE TANK
V = MEAS. TANK ON SCALE
W = SWITCH BOARD
X = HYDRAULIC PUMP
Y = LARGE TEST TABLE
Z = SMALL " "
Aa = CHAIN HOIST TEST GEAR
Bb = GAUGE TESTER
Cc = TENSILE TEST MACHINE
Dd = OIL " "
Ee = WORK BENCH
Ff = TROLLEY RAILS
Gg = GIB CRANE.

FIG. I.

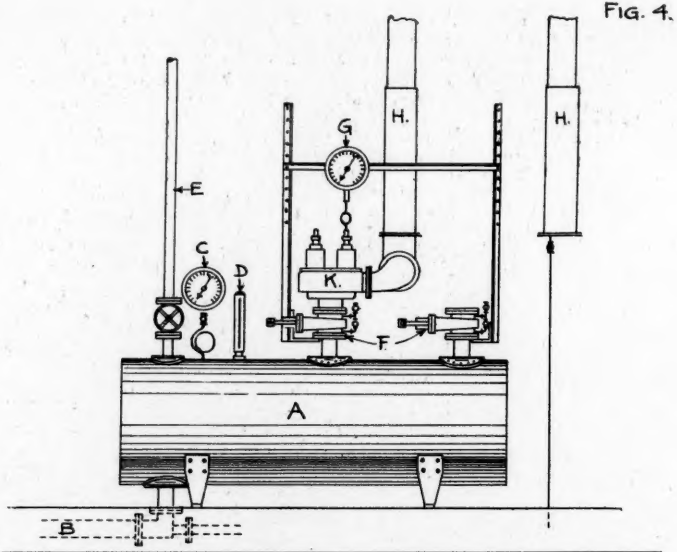


A, Boiler.
B, Smoke pipe.
C, Main steam pipe to reducing gear.
D, Feed pumps.
E, Air chamber.
F, Oil pumps.
G, Air chamber.
H. Oil heater.
K, Oil suction from supply tank.

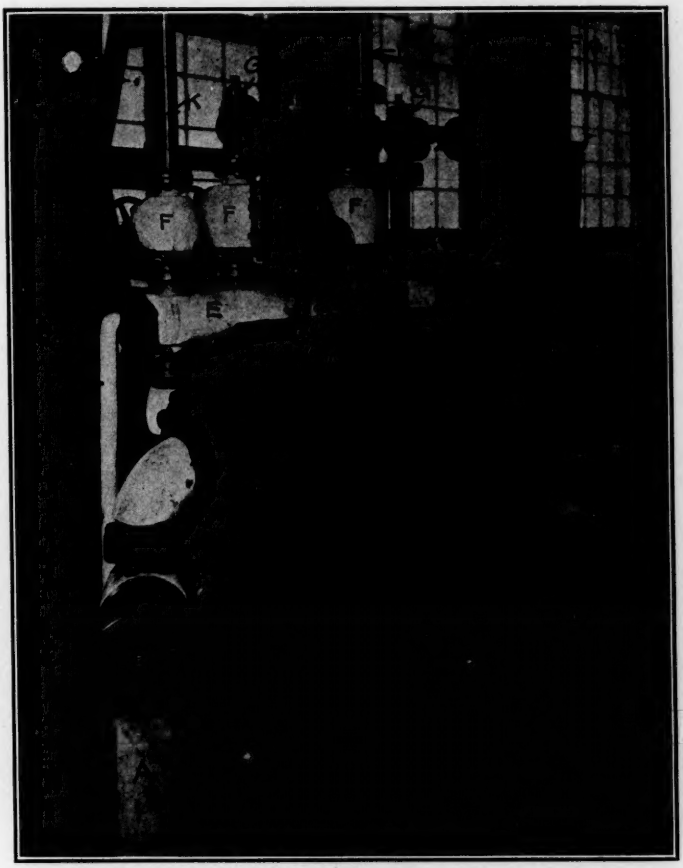


A, Steam from boiler.
B, High pressure manifold.
C, Stop valves.
D, Reducing valves.
E, Stop cross valves.
F, Relief valves.
G, Low pressure manifold.
H, Escape manifold.
K, Steam to safety-valve testing drum.
L, Steam to foundation plates and trap-test gear.
M, Drain pipes to trap.

FIG. 4.



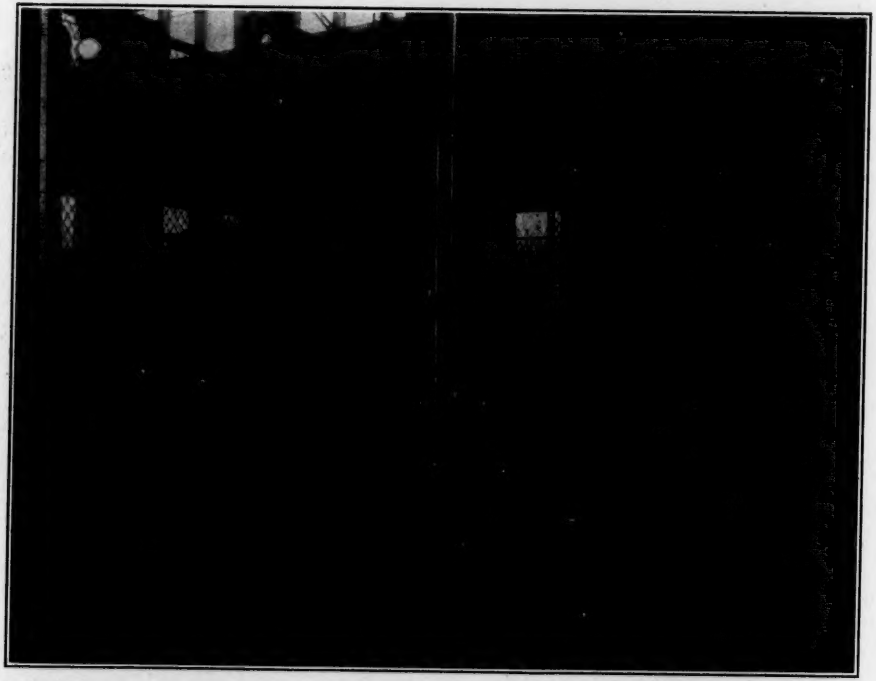
A = STEAM DRUM
B = STEAM FROM REDUCING GEAR
C = STEAM PRESSURE GAUGE.
D = THERMOMETER
E = STEAM BLEEDER
F = QU. OPEN. GATE VALVES
G = LIFT GAUGE
H = ESCAPE PIPES
K = VALVE BEING TESTED



A, Steam from reducing gear.
B, High-pressure receiver.
C, Low-pressure receiver.
D, Valve being tested.
E, Low-pressure manifold.
F, 2-inch stop valves.
G, 2-inch relief valves.
H, Escape pipe.
K, Bleeder.
L, Drain pipe.
M, Steam pipe to low-pressure manifold for setting relief valve before test.

A, Condensing tank.
B, Steam to coil.
C, Circulating-water supply.
D, Circulating-water discharge.
E, Thermometer.
F, Trap to be tested.
G, Drain to trap.
H, Measuring tank.
K, Water from trap to measuring tank.
L, Vapor pipe.
M, Water discharge.

A, Horizontal foundation plate.
B, Vertical foundation plate.
C, Steam pipe.
D, Exhaust pipe.
E, Exhaust to atmosphere.
F, Exhaust to condenser.
G, Pump being tested.
H, Trolley rail and trolley.
K, Suction hose.
L, Discharge hose.
M, Discharge manifold.
N, 5-inch stop valve.
P, 5-inch relief valves.
R, Air chamber.
S, Measuring tanks.
T, Quick-opening gate valve.
U, Discharge to measuring tank.
V, Overflow pipe.



A, Condenser.
B, Air pump.
C, Circulating water supply.
D, Circulating water discharge.
E, Exhaust pipe.
F, Air-pump suction.
G, Air-pump discharge.
H, By pass.
K, Surge tank.
L, Weighing tank.
M, Scale.
N, Discharge from weighing tank.

D, Circulating water discharge.

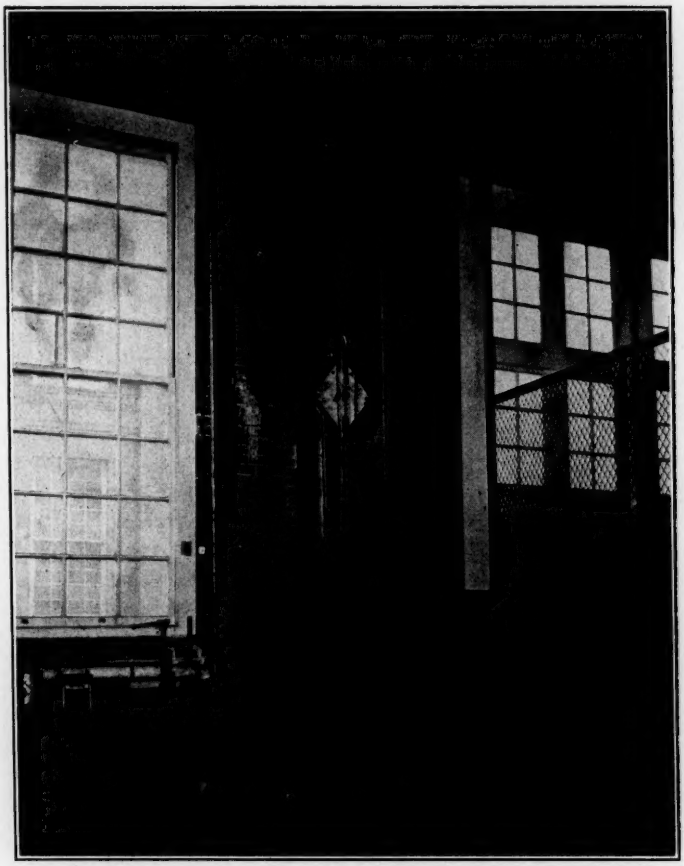
H, By pass.

N, Discharge from weighing tank.

A, Hydraulic pump.
B, Discharge manifold.
C, Stop valves.

D, Relief valves.
E, Test table (small).
F, Test table (large).

G, Steam connection.
H, Compressed-air connection.



A, Chain hoist being tested.
B, Duplex block.
C, Triplex block.
D, Single block.
E, Counterweights.
F, Anchor bolt.

tested are to be bolted. A lift gage, for registering the maximum valve lift in thousandths of an inch, is shown above the safety valve, and supported on a framework which can be adjusted to suit various different heights and positions of the valves being tested. The escape pipe, shown connected up to the safety valve, is also made adjustable by means of telescopic joints. This pipe is made of sheet iron and led out through the roof, above which it terminates with an exhaust head.

It is considered that this test unit has already well paid the cost of its installation. Several complete sets of safety valves for cruisers repairing at this navy yard have been tested, adjusted and set by this means, thereby enabling a more accurate testing and setting of the valves to be made, as well as affording better facilities for inspecting these valves under working conditions than could be had in a crowded, and usually pretty dark, boiler-room on board a ship. The expense incurred through the necessity of raising steam on each individual boiler, as has to be done when setting the safety valves on board the ship, is done away with.

For the purpose of testing and setting reducing valves a special gear has been erected, a photograph of which is shown on Fig. 5. The high-pressure steam connection to this gear is from the steam-reducing gear shown on Fig. 3 and into the small receiver shown on Fig. 5. The valve to be tested is connected to this receiver by means of a sliding joint, to provide for various different sizes of valves, and the steam from the test valve is discharged into a low-pressure receiver (C). This second receiver communicates through the manifold (E) with a set of four 2-inch relief valves and one bleeder, all of which exhaust into the atmospheric pipe. These relief valves are set ordinarily to relieve at 5, 45, 80 and 150 pounds per square inch, respectively, but can, of course, be reset to relieve at any other pressure, according to the requirement of the test, each valve within its range of pressure. Between the manifold and each relief valve, excepting one, is inserted

a stop valve, so that all valves relieving at a lower pressure than that desired for the test can be shut off. The 150-pound relief valve has been installed without a stop valve, so as to make this valve serve as the safety valve for the low-pressure side of the gear. Two pressure gages are provided, one on each receiver, to enable the operator at all times to check the performance of the valve being tested. A simple arrangement for testing steam traps is shown on Fig. 6. It consists of the condensing tank (A) containing a copper pipe coil, into which steam is led from the main steam pipe. As the main steam connects to the steam-pressure reducing gear, any desired pressure up to 400 pounds per square inch can be obtained in the condensing coil. The circulating water for the condensing tank enters at the bottom and discharges through any one of the four overflow connections shown, so that the amount of condensation can be regulated within certain limits by having a greater or smaller portion of the steam coil submerged. A thermometer is inserted in the circulating-water discharge to provide an indicator by which to regulate the supply of circulating water and prevent a useless waste of same. The trap to be tested is shown resting on a bracket and ready to be connected to the lower end of the steam coil in the tank. The discharge from the trap goes into a tank (H) provided with a graduated gage glass so that the amount of water discharged can be easily determined. This tank is made a closed tank to prevent any splashing of water when the trap discharges. An open stand pipe (L) forms the required exit for vapor. The water from this measuring tank, as well as the circulating-water discharge, is led to the sewer.

On Fig. 7 will be seen the foundation plates, horizontal and vertical, which have been provided for setting up, for running test, machinery such as launch engines and other steam engines of 300 H.P. or less, gasoline engines, pumps of all kinds, electric motors and generators. The horizontal plate is 4 feet by 12 feet, in two sections, and bolted down on top of a concrete tank, the use of which is explained further on in this

article. The vertical, or angle plate, is 3 feet wide by 6 feet high, and rests on a special foundation plate, on which it is adjustable by means of screws in two directions, similar to the tool holder on a lathe. This was done to facilitate the fitting up of discharge connections from pumps being tested and supported on the vertical foundation plate. Both plates are provided with tee slots for receiving the heads of machinery holding-down bolts. These slots run in two directions, at right angles, on both the plates. At about midlength of the horizontal plate and some 7 feet above same are seen the ends of the steam and exhaust pipes to which the engine or pump under test is to be connected. The steam pipe connects to the steam-reducing gear seen in Fig. 3, and can be made to deliver steam at any pressure up to 400 pounds per square inch. The exhaust pipe is connected at will to either the atmosphere or the condenser.

The condensing plant, shown in Fig. 8, consists of a surface condenser, of 1,400 square feet cooling surface, removed from the former U.S.S. *Davis*. A duplex, double-acting air pump, steam-driven, 5-inch by 5¼-inch by 3¼-inch, has been made up by the navy yard, using the steam end from the old circulating engine of the *Davis*, and manufacturing a new air end. As the condenser capacity is away in excess of the requirements of the station, it is intended, at some future time, to cut off part of the condenser, in order to provide a better proportion between condenser capacity and air pump, and so obtain a better vacuum than is possible at present. A platform scale and tank for weighing the condensed steam is provided. This, and the somewhat smaller surge tank, above it, are plainly seen in the figure. The surge tank was provided so that the discharge from the air pump could be taken up and stored in this tank while the weighing tank was shut off and the contents of this latter being weighed and emptied. Circulating water for the condenser is obtained from the yard system. In order to prevent a waste of circulating water, a thermometer has been installed in the discharge line and a

throttle valve in the supply line, so that the proper rate of flow can be easily determined and maintained.

At one end of the horizontal foundation plates, seen on Fig. 7, there has been space reserved for the installation of a 300 H.P. hydraulic brake. This brake has been provided to enable horsepower tests of engines and motors to be made. Its location is indicated in Fig. 1 at (P). (W) on the same figure shows the location of a switchboard, in close proximity to the foundation plates and the brake. This switchboard has connections to a motor generator set, installed in the electric laboratory, so that tests on any ship's motors, up to 60 H.P., can be made. The motor generator set is of 50,000 watts capacity, and capable of delivering direct current at any voltage from 50 to 150. The motor end consists of a three-phase, 60-cycle induction motor taking current at 2,300 volts. The generator is of the direct-current compound-wound type. Its normal rating is 50,000 watts at 125 volts, but a wide variation of voltage, by means of field rheostat, has been provided for, so that the exact voltage required in the test can always be provided. A local control of the motor generator has been provided for on the panel in the mechanical test station.

The apparatus for testing pumps, which is shown on Fig. 7, consists of four main units: foundation plates, suction tank, pressure-regulating gear and measuring tanks. The foundation plates have been described previously. The suction tank is a concrete tank 4 feet by 14 feet by 2 feet 10 inches deep, installed directly below the foundation plates. Its top is level with the floor and forms the foundation for the cast plates. Along one side of the tank there is an opening, about 7 inches wide, left between the edge of the tank and that of the cast-iron plate, so as to facilitate the lowering of a suction pipe into the tank at any place where considered most convenient for the test. The discharge from the pump being tested is carried into the manifold (M) shown. On top of this manifold will be seen four 5-inch stop valves, each one connecting to a 5-inch relief valve. The ranges of relief pressure of these

valves are: one 100–150, one 150–250, one 250–350, and one 350–500 pounds per square inch. By opening up the discharge passage through the proper relief and setting this valve to the pressure it is desired to pump against, it will be seen that a test can be had at any discharge pressure from 100 to 500 pounds per square inch. It should be added that a large air chamber of 2,300 cubic inches is connected to the manifold, in order to provide a steady pressure to pump against. The figure shows the lower end only of this air chamber (R) and the pipe connecting it to the manifold. The discharge nozzles of all four relief valves connect into one manifold, from where the water is directed to one measuring tank or the other by means of two 5-inch quick-opening gate valves (T). Each measuring tank (S) holds 500 gallons up to its overflow. The drain is by means of an 8-inch sluice valve in the bottom of each tank, which discharges the water back into the suction tank. As each tank overflows into the other tank there is no water being passed out during a test without being measured, provided care is taken to shift the discharge as soon as one tank has been filled and commences to overflow, and to promptly drain the full tank. A prolonged capacity test on a pump can thus be had by alternately shifting the discharge between the two measuring tanks and draining same, the limit being set by the capacity of drain connection, which has proven to be about 650 gallons per minute. Both suction and measuring tanks are provided with emergency overflows, open at all times, in order to prevent a flooding of the premises. Several pumps of varying capacities have been successfully tested in this set. A remarkably steady discharge pressure is maintained in the line, on account of the large air chamber. The pressure fluctuations generally average about 5 pounds per square inch.

For cold-water pressure tests there are provided one pump and two test tables, all shown on Fig. 9. The pump is a 6-inch by 2¼-inch by 7-inch duplex, double-acting pump, steam or air driven. The two steam cylinders were obtained from the

main feed pump of the former U.S.S. *Davis,* and the water end was made new by the navy yard. It is of cast iron with brass liners, monel-metal valve seats and steel ball valves. It is arranged to obtain water pressures up to 1,000 pounds per square inch, but the pump cylinders have been tested to 1,500 pounds. By means of the seven 1¼-inch relief valves shown on the figure, each one, excepting the last one, connected through a stop valve to the discharge manifold of the pump, any test pressure can be obtained by opening the stop valve connecting to the proper relief valve and setting this relief valve to the exact pressure required. The pressure ranges of the relief valves are as follows: 100–200, 200–300, 300–400, 400–550, 550–750 and 750–1,000 pounds per square inch. The last valve is installed without a stop valve and is set permanently at 1,000 pounds per square inch, so as to serve as safety valve for the whole system. A large air chamber is connected to the pump-discharge chamber and serves its purpose of steadying the pressure very well. From the pump manifold there are connections made to two test tables, both shown on Fig. 8. The one in the rear on the figure has four pressure connections, arranged to accommodate valves and fittings of from ⅜-inch to 4½-inch size. On the two middle connections the holding-down device consists of yoke and handwheel located above the fitting being tested, whereas the two outer connections have been arranged with the handwheel for clamping below. The former is the simpler and most easily operated rig, but the latter has the advantage of leaving the test piece free and accessible on all sides for inspection, a point which is considered of great importance in testing valves. The test table shown in front on the figure has two pressure connections, to accommodate valves and fittings of from 5-inch to 15-inch size, inclusive. The holding-down gear consists of clamps on top of the table, held down by bolts working in slots cast in the table top. This leaves the test piece free for inspection, as shown on the figure. Connections are also made from the steam lines and compressed-air lines to the

hydraulic pipes connecting to these test tables, so that an air or steam-pressure test can be had when required.

In Fig. 10 will be seen an apparatus which has been erected for the purpose of giving a rapid method of applying test loads to chain hoists. It has a capacity up to 8,000 pounds, so that hoists up to three tons capacity can be tested to a 25 per cent. overload. The figure is very nearly self-explanatory. It is evident that the load on the hoist being tested must equal the pull on the anchor bolt minus the counterweight (E), if the friction in block (D) is disregarded. A careful calibration of the set has been made to determine the pull on the anchor bolt for a load at (E) of from 50 to 1,200 pounds, both when the load was resting and when it was being hoisted. The results obtained determine the load curve for the apparatus, which will give the weight required at (E) for any load desired on the hoist to be tested. The device used during calibration is not shown on the figure. After this apparatus had been used for a short time it was found more expedient, as well as safer, to suspend the two blocks (B) and (C) above the chain hoist, the hook of the latter connecting to the anchor bolt. This alteration of the apparatus has been made and a recalibration done, but the description here given is for the original gear, so that reference could be had to the photograph.

One dead-weight gage tester with a capacity up to 5,000 pounds per square inch, one tensile-test machine and one oil-testing machine complete the present stationary equipment of the laboratory. The location of these units is shown on the general plan, Fig. 1. It should be mentioned here that there have been added some special fittings to the tensile-test machine whereby this machine can be adapted to making tests on helical springs, giving the deflection at any carrying capacity.

The facilities for handling machinery in the test station consist of two trolley rails and one column gib crane. The trolley rails are designed for a load of 6,000 pounds and are mainly intended for handling heavy machinery between the shipping station and the foundation plates, as will be apparent from an

examination of Fig. 1. The trolley rail is extended far enough to enable a transfer of the load to be made to the gib crane when such is necessary. The gib crane is supported on one of the building columns, and has a maximum capacity of 2,000 pounds. It reaches over the test apparatus for safety valves and that for reducing valves, as well as over both the hydraulic tables, and its mean functions are those of handling the valves and fittings being tested in these places.

BLOWER PERFORMANCE,

AS AFFECTED BY SIZE AND SPEED OF BLOWER AND DENSITY OF AIR MOVED.

By Wm. L. DeBaufre, Mechanical Engineer.

The following article was written primarily to explain the basis for reducing the data from tests on forced-draft blowers at the Naval Engineering Experiment Station to standard conditions. The standard conditions are assumed to obtain with an air density of 0.07492 pound per cubic foot, corresponding to dry air containing the normal amount of CO_2, 0.04 per cent. by volume, at a temperature of 70 degrees F. and under a barometric pressure of 29.92 inches mercury. The article has been extended, however, to show how to derive from the results under standard conditions, the performance to be expected with a different air density due to humidity, temperature or pressure variation, with a different speed of the blower, and also with another blower geometrically similar to the one tested.

FORCED-DRAFT SYSTEMS ON SHIPBOARD.

To serve as a basis for the derivation of the desired relations, consider the forced-draft systems as installed on a battleship, Fig. 1, and on a destroyer, Fig. 2. In both cases the blower D draws air through the ventilator V from the atmosphere A and discharges the air into the closed fireroom F, from which it escapes through the boilers B and stack S to the atmosphere again. The essential difference between the two systems is that encased horizontal blowers, electric motor or steam turbine driven are installed on battleships, while on destroyers are mounted open vertical blowers driven by steam turbines

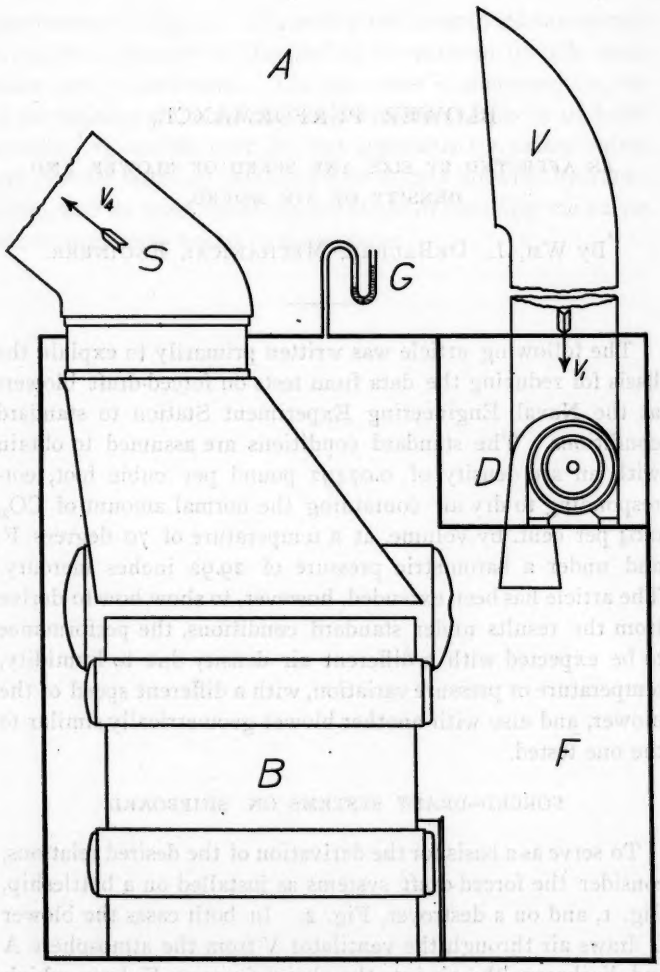


FIG. 1.

suspended beneath. The static pressure above atmospheric pressure produced in the fireroom by the blower is indicated by the differential water gage G.

In forcing air through such a system frictional resistance

is encountered, part in the blower itself and part in the ventilators, boilers and stack. Also, the velocity with which the air leaves the stack represents a certain amount of kinetic energy which has been added to the air, originally at rest in the atmosphere. The energy necessary to overcome the frictional resistances encountered in the system and to impart the velocity at leaving, is supplied mainly by the blower, although a portion comes from the stack "draft." We will neglect for the present the "draft," and assume that all of the required energy is supplied by the blower. The fundamental equation, therefore, is that the energy required to move the air through the system is equal to the energy supplied by the blower wheel, say per pound of air moved.

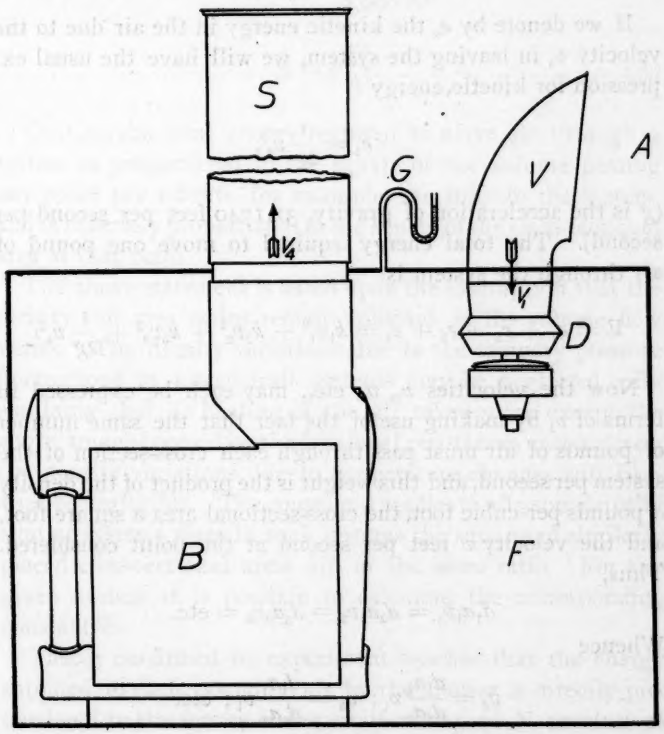


FIG. 2.

DERIVATION OF FORMULAS.

Experiment shows that the number of foot-pounds of energy consumed by frictional resistance per pound of air is proportional to the square of the velocity measured in feet per second at the point considered. Expressed algebraically,

$$e_1 = k_1 v_1^2,$$

where e_1 is the energy consumed per pound of air, v_1 is the air velocity, and k_1 is some constant. The above relation might apply, for example, to the inlet of the system; but for other points we will likewise have

$$e_2 = k_2 v_2^2;\ e_3 = k_3 v_3^2, \text{ etc.}$$

If we denote by e_4 the kinetic energy in the air due to the velocity v_4 in leaving the system, we will have the usual expression for kinetic energy

$$e_4 = \frac{1}{2g} v_4^2,$$

(g is the acceleration of gravity, 32.1740 feet per second per second). The total energy required to move one pound of air through the system is

$$E = e_1 + e_2 + e_3 + e_4 = k_1 v_1^2 + k_2 v_2^2 + k_3 v_3^2 + \frac{1}{2g} v_4^2.$$

Now the velocities v_2, v_3, etc., may each be expressed in terms of v_1 by making use of the fact that the same number of pounds of air must pass through each cross-section of the system per second, and this weight is the product of the density d pounds per cubic foot, the cross-sectional area a square foot, and the velocity v feet per second at the point considered. Thus,

$$d_1 a_1 v_1 = d_2 a_2 v_2 = d_3 a_3 v_3 = \text{etc.}$$

Whence

$$v_2 = \frac{d_1 a_1}{d_2 a_2} v_1;\ v_3 = \frac{d_1 a_1}{d_3 a_3} v_1;\ \text{etc.}$$

Substituting in the above expression for the total energy required, we obtain

$$E = \left[k_1 + k_2\left(\frac{d_1a_1}{d_2a_2}\right)^2 + k_3\left(\frac{d_1a_1}{d_3a_3}\right)^2 + \frac{1}{2g}\left(\frac{d_1a_1}{d_4a_4}\right)^2\right]v_1^2 = kv_1^2.$$

If Q represents the volume of air passing into the system in cubic feet per minute, then

$$Q = 60Av_1,$$

where A is the entrance area in square feet.

Therefore,

$$E = kv_1^2 = \frac{k}{3{,}600}\frac{Q^2}{A^2}.$$

Or,

$$E = k\frac{Q^2}{A^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

That is, the total energy required to move air through a system is proportional to the square of the volume passing any point per minute, for example, the inlet to the system, and is inversely proportional to the square of the cross-sectional area at that point.

The above statement is based upon the assumption that the density and area ratios remain constant as the volume flow varies. The density variations due to the ordinary pressure fluctuations in forced-draft systems may be neglected. By assuming that no heating of the air takes place except the slight amount caused by the frictional resistances encountered, the density variations due to temperature changes will likewise be negligible. The constant k applies to all geometrically similar systems since in such systems the squares of similarly placed cross-sectional areas are in the same ratio. For any given system it is possible to calculate the corresponding constant k.

Theory confirmed by experiment teaches that the energy supplied to each pound of air by the blower is directly proportional to the square of the rotational speed N revolutions

per minute and also to the square of the wheel diameter D feet for geometrically similar blowers, provided the volume flow Q cubic feet per minute varies directly as N and as the cube of D. That is,

$$E = CD^2N^2 \text{ provided } Q = k_1D^3N. \quad . \quad . \quad . \quad (B)$$

The truth of the relations will be made evident by the following explanations.

Refer to Fig. 3, A, B and C, representing wheels with radial, backward-curved and forward-curved vanes, respectively. In each case the velocity v of the air leaving the wheel is the geometric sum of the circumferential velocity $\frac{\pi DN}{60}$ feet per second and the flow velocity $\frac{Q}{60\,\pi DB \sin\beta}$ feet per second, where B is the breadth of the wheel at its outer circumference and β is the angle between the tip of the blade and the tangent at the circumference. The entering velocity is $v_1 = \frac{Q}{60A_1}$ feet per second, where A_1 square feet is the area at entrance. The kinetic energy added to each pound of air is $\frac{v^2}{2g} = \frac{v_1^2}{2g}$ foot pounds. Now, one component of v is proportional to the speed N; if Q were proportional to N the other component of v and also v_1 would be proportional to N_1, and the kinetic energy added would be proportional to N^2. Likewise, one component v is proportional to D; if Q were proportional to D^3, the other component of v and also v_1 would be proportional to D, in geometrically similar wheels, and the kinetic energy added would be proportional to D^2. Hence, relations B are true for the kinetic energy added to the air by the wheel. (A properly designed casing converts part of this kinetic energy into potential energy.)

With whatever form of blade is used the air is pressed by centrifugal force from the inner to the outer circumference of the wheel. The static pressure at the inner circumference corresponds to the centrifugal force $\frac{D_1}{2g}\left(\frac{2\pi N}{60}\right)^2$ pounds, and at

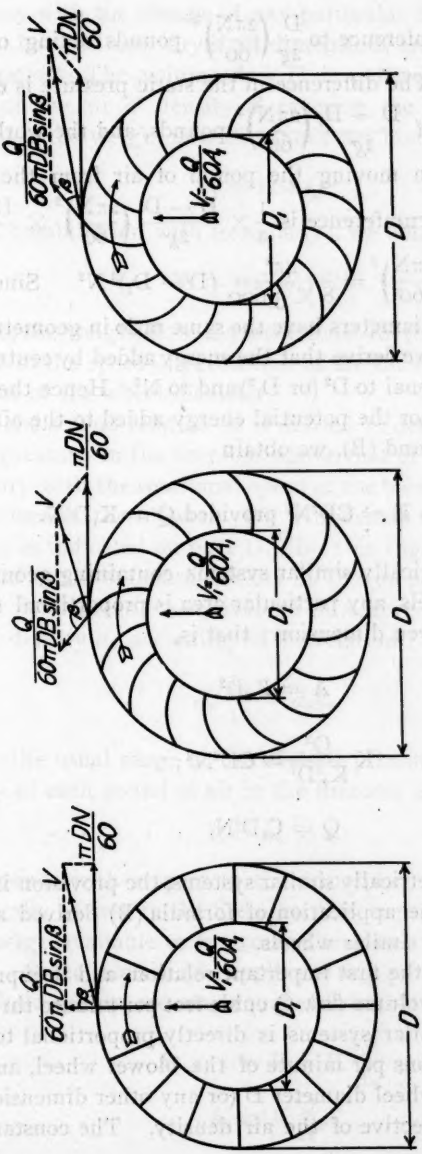


FIG. 3.

the outer circumference to $\frac{D}{2g}\left(\frac{2\pi N}{60}\right)^2$ pounds acting on one pound of air. The difference in the static pressure is equivalent to a force of $\frac{D - D_1}{2g}\left(\frac{2\pi N}{60}\right)^2$ pounds, and the work done by this force in moving the pound of air from the inner to the outer circumference is $\frac{1}{2} \times \frac{D - D_1}{2g}\left(\frac{2\pi N}{60}\right)^2 \times \frac{D - D_1}{2}$ $= \left(\frac{D - D_1}{8g}\right)^2 \left(\frac{2\pi N}{60}\right)^2 = \frac{4\pi^2}{8 \times 3{,}600}(D - D_1)^2 N^2$. Since the inner and outer diameters have the same ratio in geometrically similar wheels, we derive that the energy added by centrifugal force is proportional to D^2 (or D_1^2) and to N^2. Hence the truth of relations (B) for the potential energy added to the air.

Equating (A) and (B), we obtain

$$K\frac{Q^2}{A^2} = E = CD^2N^2 \text{ provided } Q = K_1D^3N.$$

Now, in geometrically similar systems containing geometrically similar wheels, any particular area is proportional to the square of any given dimension; that is,

$$A = K_2D^2.$$

Hence,

$$K\frac{Q^2}{K_2^2D^4} = CD^2N^2;$$

Whence,

$$Q = C_1D^3N. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Thus, in geometrically similar systems, the provision is met that restricted the application of formula (B) derived above for geometrically similar wheels.

Formula (1) is the first important relation, and it expresses the fact that the volume flow Q cubic feet per minute through geometrically similar systems is directly proportional to the speed N revolutions per minute of the blower wheel, and to the cube of the wheel diameter D (or any other dimension of the system) irrespective of the air density. The constant C_1

will vary with the change of any particular dimension of the system, but will not vary if all dimensions are changed in the same ratio. The volume flow Q is generally expressed in terms of the air of density d_a entering the blower. If W represents the weight flow in pounds per hour, we have

$$W = 60 d_a Q; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (C)$$

and by combination with formula (1), we obtain

$$W = 60 C_1 d_a D^3 N. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

That is, the weight flow is proportional to the air density as well as to the speed of revolution of the blower wheel and to the cube of the wheel diameter.

For the second relation it is desired to determine how the static pressure in the fireroom above that of the atmosphere will vary with the rotational speed of the blower and with the dimensions of the blower. We will denote this pressure difference as indicated on gage G, Fig. 1 or Fig. 2, by h_w inches of water at the standard density d_w (62.30 pounds per cubic foot at 70 degrees F.) equivalent to h_a feet of air at atmospheric density d_a, according to the relation

$$d_w \frac{h_w}{12} = d_a h_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (D)$$

For the usual range of forced-draft pressures the potential energy of each pound of air in the fireroom is equal to

$$E = h_a \times 1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (E)$$

foot-pounds with respect to the atmosphere. As the air is practically at rest in the fireroom, this expression represents the energy available to overcome the losses in the boiler and stack. Since relation (A) applies to any part of as well as to the whole system, we may equate (A) and (E), obtaining

$$h_a \times 1 = E = K \frac{Q^2}{A_2},$$

$$h_a = K \frac{Q^2}{A^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (F)$$

Thus, in geometrically similar systems the head in feet of air is proportional to the square of the volume flow resulting therefrom, and inversely proportional to the square of any area (or to the fourth power of any dimension). Replacing the head in feet of air by the equivalent static pressure in inches of water, we obtain

$$h_w = \frac{12K}{d_w} d_a \frac{Q^2}{A^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (F\text{-}1)$$

The static pressure as indicated on the differential water gage will therefore vary with the air density. Combining formula (1) with formulas (F) and (F-1) respectively, and remembering that $A = K_2 D^2_1$, we obtain

$$h_a = C_2 D^2 N^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and

$$h_w = \frac{12C_2}{d_w} d_a D^2 N^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

The pressure in the fireroom is thus proportional to the square of the speed of the blower, and in geometrically similar systems is proportional to the square of the wheel diameter.

Now let us determine how the power required to drive the blower varies with its speed and size. Formula (B) $E = CD^2N^2$, gives the energy imparted to each pound of air by the blower. The number of pounds of air receiving this energy per hour is given by formula (1a), namely,

$$W = 60\, C_1 d_a D^3 N.$$

Hence, the number of foot-pounds required per hour to move the air is

$$WE = 60\, CC_1 d_a D^5 N^3.$$

The power lost in the bearings is but a small proportion of the total power expended, and therefore a small error only is introduced by assuming the bearing loss p to be proportional to the same powers of the same quantities. This assumption is not as far from the truth as it might at first thought appear. We may write

$$p = k d_a D^5 N^3.$$

Whence,

$$WE + p = 60\,CC_1 d_a D^5 N^3 + k d_a D^5 N^3.$$

Or,

$$P = C_3 d_a D^5 N^3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For geometrically similar systems the power input is therefore within close limits proportional to the air density, to the fifth power of the wheel diameter, and to the cube of the rotational speed. The value of the constant C_3 will depend upon the relative dimensions of the blower and system, and also upon the unit used to measure the power, whether foot-pounds per hour or shaft horsepower.

The efficiency of a blower is equal to the effective work done divided by the total energy expended in the same time in driving the blower. The effective work done by a forced-draft blower on board ship is measured by the potential energy in the air in the fireroom available to force the air through the boiler and stack. Since by formula (E) there are h_a foot-pounds of potential energy in each of the W pounds of air forced per hour into the fireroom, we obtain from formulas (1a) and (2) the effective work done per hour to be

$$Wh_a = 60\,C_1 C_2 d_a D^5 N^3.$$

The energy expended per hour in driving the blower is, from formula (3),

$$P = C_3 d_a D^5 N^3.$$

Hence, for the efficiency we obtain

$$e = \frac{Wh_a}{P} = \frac{60\,C_1 C_2 d_a D^5 N^3}{C_3 d_a D^5 N^3} = \frac{60\,C_1 C_2}{C_3}.$$

Or,

$$e = C_4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

That is, when discharging air through geometrically similar systems the efficiency of geometrically similar blowers will be the same, no matter how the rotational speed of the blower may vary with the resultant variation in volume flow nor even how much the density of the air may change. This relation

may be called the fourth important relation, and experiment shows that it holds true to a remarkable degree.

The above formulas have been derived as for blowers installed in particular systems; but the systems may be considered imaginary and the formulas applied to obtain from test data for a given blower under particular conditions, the performance of similar blowers under other conditions. It is necessary to remember, however, that any one formula cannot be applied unless all the remaining formulas are considered true at the same time. Thus, the volume flow cannot be considered as varying directly with the speed unless the head is considered to be proportional to the square of the speed and the power to the cube of the speed, all at a constant efficiency.

The succeeding results are based on these formulas. The important ones, showing the variation of the discharge, the head, the power, and the efficiency with the air density, and the size and speed of the blower, are tabulated below for reference. Many combinations of these primary formulas might be made. The symbols are explained as referring to the data obtained by testing a particular blower. The formulas are:

$$Q = C_1 D^3 N \text{ or } Q_2 = \frac{D_2^3 N_2}{D_1^3 N_1} Q_1 \quad . \quad . \quad . \quad . \quad (1)$$

$$W = 60\, C_1 d_a D^3 N \text{ or } W_2 = \frac{d_{a_2} D_2^3 N_2}{d_{a_1} D_1^2 N_1} W_1 \quad . \quad . \quad (1a)$$

$$h_a = C_2 D^2 N^2 \text{ or } h_{a_2} = \frac{D_2^2 N_2^2}{D_1^2 N_1^2} h_{a_1} \quad . \quad . \quad . \quad (2)$$

$$h_w = \frac{12 C_2}{d_w} d_a D^2 N^2 \text{ or } h_{w_2} = \frac{D_{a_2} D_2^2 N_2^2}{D_{a_1} D_1^2 N_1^2} h_{w_1} \quad . \quad . \quad (2a)$$

$$P = C_3 d_a D^5 N^3 \text{ or } P_2 = \frac{d_{a_2} D_2^5 N_2^3}{d_{a_1} D_1^5 N_1^3} P_1 \quad . \quad . \quad . \quad (3)$$

$$e = C_4 \text{ or } e_2 = e_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which

D = diameter in feet of the wheel in the blower under test;
N = rotational speed in revolutions per minute during the run;

d_a = density of the air in pounds per cubic foot at the inlet to the blower during the run;

d_w = standard density of water, 62.30 pounds per cubic foot at 70 degrees F.;

Q = volume flow of air during the run expressed in cubic feet per minute of density d_a;

W = weight flow of air during the run in pounds per hour;

h_a = effective static head during the run expressed in feet of air of density d_a;

h_w = effective static pressure during the run reduced to the standard water density d_w;

P = power to drive the blower during the run expressed either in foot pounds per hour or in shaft horsepower;

and

e = efficiency of the blower calculated from the test data for the run conditions.

The constants C_1, C_2 and C_3 will correspond to C_4, but they will also vary with the units employed to measure the several quantities.

REDUCTION TO STANDARD AIR DENSITY.

Standard conditions would correspond to the circulation of air of the standard density, 0.07492 pound per cubic foot, instead of the test density of d_a pounds per cubic foot; the speed, however, to be the same under standard as under test conditions. While these formulas serve for speed as well as for density reductions, as will be explained later, it is now assumed that the speed of the blower has been regulated very closely to the desired value during the test run, but that it was necessary to use air appreciably different from standard density.

From formula (1) we see that the volume flow is not affected by air density, so that no correction is necessary for the number of cubic feet of air moved per minute. The weight flow, as indicated by (1a), is affected by the density; but instead of applying formula (1a) directly, the simplest way to determine

the number of pounds per hour under standard conditions is to multiply the volume flow in cubic feet per minute by 60 minutes per hour and by 0.07492 pound per cubic foot (the standard air density).

From formula (2) it is evident that the effective static head in feet of air would be the same under standard and under test conditions; but the effective static pressure expressed in inches of water would not be the same, as indicated by formula (2a). The effective static pressure under standard conditions would equal the effective static head in feet of air calculated from the run data multiplied by 12 inches per foot and by the standard air density and divided by the standard water density. This factor is accordingly $\frac{12 \times 0.07492}{62.30} = 0.014431$.

The shaft horsepower under standard conditions may be obtained from the power measured during the run by multiplying by the ratio of standard to test air density, or it may be directly calculated from the standard weight flow and head. The efficiency will be the same under standard as under test conditions.

The methods of applying these reductions are indicated by the following items abstracted from a table of test data and results for a particular blower. No reduction is considered necessary in the number of pounds of steam per horsepower per hour, as the variation in this quantity is due only to the change in the steam economy of the driving turbine between the standard and test horsepowers, which will generally be slight.

Static pressure at blower suction, inches water gage, . . 0
From test data corrected to a water density of 62.30 pounds per cubic foot.

Static pressure at blower discharge, inches water gage, 3.471
From test data corrected to a water density of 62.30 pounds per cubic foot.

Air density at suction, pounds per cubic foot, . . 0.07313
Calculated from test data.

Effective static head, feet air, ... 246.41

From $\frac{(3.471 - 0) \times 62.30}{12 \times 0.07313}$.

Effective static pressure under standard conditions, inches water, ... 3.556

From $\frac{246.41 \times 0.07492 \times 12}{62.30}$.

Air discharged under test conditions per hour, pounds, 68,382

Calculated from the test data.

Air discharged per minute, cubic feet, ... 15,585

From $\frac{68,382}{60 \times 0.07313}$.

Air discharged under standard conditions per hour, pounds, ... 70,058

From $15,585 \times 0.07492 \times 60$.

Work done under test conditions per hour, 1,000,000 foot pounds, ... 16.850

From $\frac{68,382 \times 246.41}{1,000,000}$.

Air horsepower under test conditions, ... 8.51

From $\frac{16.850}{1.98}$, where 1.98 million foot-pounds per hour is equivalent to one horsepower.

Shaft horsepower under test conditions, . . . 17.80

From test data.

Shaft horsepower under standard conditions, . . 18.24

From $17.80 \times \frac{0.07492}{0.07313}$.

Fan efficiency, per cent., ... 47.81

From $\frac{8.51}{17.80}$.

Steam used per hour, pounds, ... 1,386
From test data.

Steam economy per air horsepower hour, pounds, . 162.9
From $\frac{1,386}{8.51}$.

TESTING BLOWERS.

Forced-draft blowers with casings, as installed on battleships, are tested at the Naval Engineering Experiment Station by discharging the air into a cylindrical galvanized-iron chamber, 10 feet long by 10 feet diameter, which accordingly represents the fireroom as indicated in Fig. 4. The air escapes from this chamber C through nozzle N; the energy consumed in setting the air in the nozzle in motion and in overcoming the frictional resistances therein corresponds to the energy consumed in the boilers and stack on board ship. By proper measurements the items in the above table are obtained, the weight flow being calculated from the velocity head of the air as it escapes from the nozzle. The weight flow may also be determined from venturi meter or pitot-tube readings in the discharge duct from the chamber. In these tests the effective static pressure is taken as the difference between the laboratory air pressure and the average of several pressures measured in the discharge chamber. Hence, in applying the test results to conditions after installation on board ship, allowance must be made for the frictional losses in the ventilator ducts to obtain the air pressure at the boiler fronts.

It is more convenient to test destroyer fireroom blowers, which have no casing, by connecting the suction rather than the discharge duct to the chamber C, as shown in Fig. 5. The air is drawn into chamber C through nozzle N and venturi meter M by the suction of blower B, in this case driven by an electric dynamometer. The air in the chamber then represents the "atmosphere," and the whole laboratory represents the "fireroom." The effective static pressure is measured between the chamber and the laboratory, and this air pressure

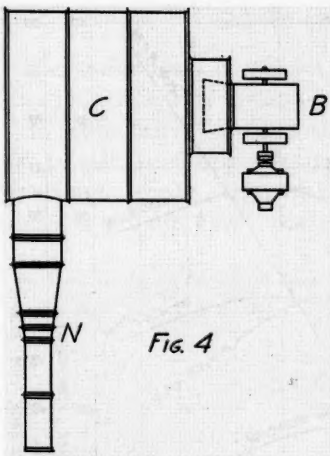


FIG. 4

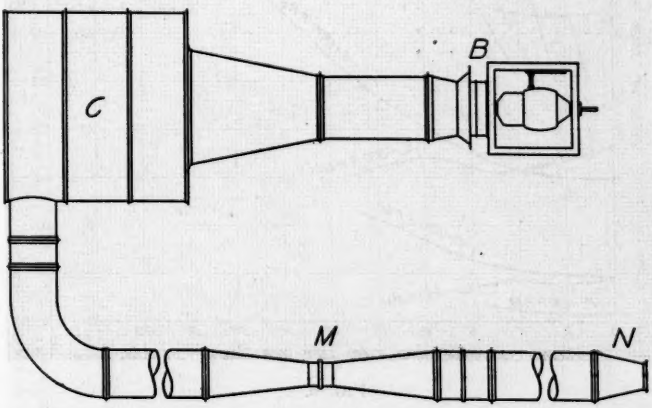


FIG. 5.

will be obtained at the boiler front because the suction duct between the chamber and the blower is made to correspond to the ventilator and duct on a destroyer.

When additional space becomes available at the Naval Engineering Experiment Station by the erection of a new building, it is hoped to construct a large airtight room especially adapted for blower tests and investigations.

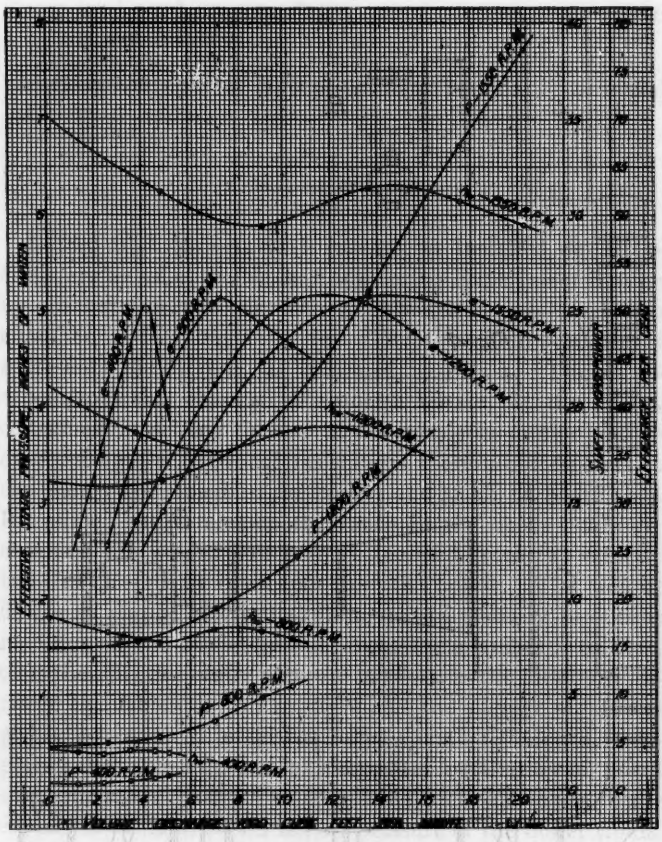


FIG. 6.

By making a number of runs in the above manner at rated speed and at other constant speeds less than rated speed, there are obtained a series of curves as given in Fig. 6, showing the variation of effective static pressure, of shaft horsepower, and of efficiency with the volume flow at standard air density. From these curves may be determined the speed and horsepower of the particular blower tested, which will give the volume flow and air pressure at the boiler fronts desired.

SPEED REDUCTION.

The performance of the blower at speeds intermediate to those plotted in Fig. 6 may be obtained by means of formulas (1) to (4) inclusive. In fact, the characteristics at any speed may be calculated to a close approximation from the test data for a single speed. Thus, in Fig. 7, the curves at 800 revolutions per minute have been calculated from those at 1,550

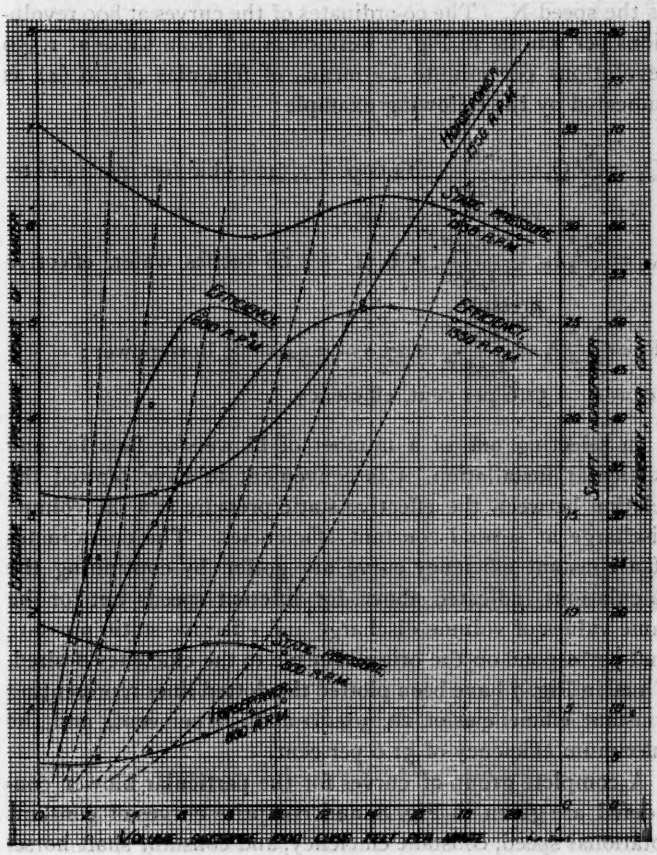


FIG. 7.

revolutions per minute, and may be compared with the points marked thereon, obtained by actual measurement at 800 revolutions per minute.

It should be emphasized that in making the reductions for speed any one of the formulas (1) to (4) holds true only when all the others hold true. That is, the effective static pressure h_w (at constant air density) does not vary as the square of the speed unless the volume flow Q at the same time varies directly as the speed N. The co-ordinates of the curves at 800 revolutions per minute in Fig. 7 were obtained from those at 1,550 revolutions per minute by means of formulas (1) to (4), as indicated by the following example:

$$Q_2 = \frac{N_2}{N_1} Q_1 = \frac{800}{1{,}550} \times 14{,}600 = 7{,}540 \text{ cubic feet per minute, volume discharge;}$$

$$h_{w_2} = \frac{N_2^2}{N_1^2} h_{w_1} = \frac{800^2}{1{,}550^2} \times 6.31 = 1.68 \text{ inches water, effective pressure;}$$

$$P_2 = \frac{N_2^3}{N_1^3} P_1 = \frac{800^3}{1{,}550^3} \times 27.9 = 3.84 \text{ shaft horsepower;}$$

$$e_2 = e_1 = 51.6 \text{ per cent. efficiency.}$$

In selecting one of a number of blowers or fans for any particular installation, it is convenient to have plotted the relation between the effective static pressure and the volume discharge at several constant efficiencies. The co-ordinates of a constant efficiency curve are gotten by continuing the calculations indicated above, for different speeds but for the same values of volume discharge and effective static pressure that correspond on the original curve to the efficiency desired. Thus, in Fig. 7 have been plotted the dotted lines representing constant efficiencies of 20, 30, 40 and 50 per cent., and the maximum efficiency of 51.6 per cent.

A complete group of curves for the particular blower considered are given in Fig. 8, and comprise curves of constant rotational speed, constant efficiency, and constant shaft horsepower at various effective static pressures and volume dis-

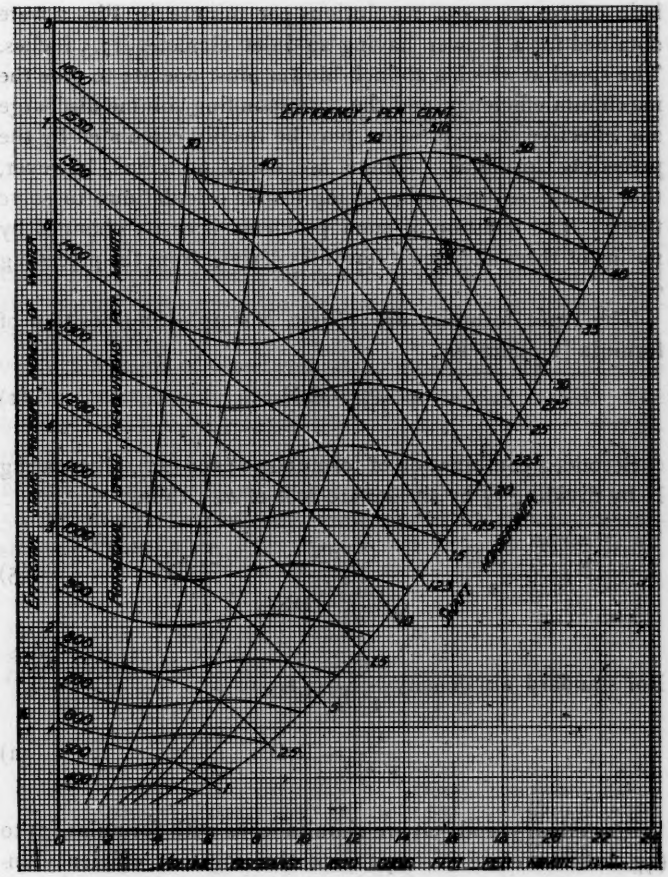


FIG. 8.

charges. These curves were obtained from those in Fig. 6 by cross-plotting and by the use of the above formulas.

GEOMETRICALLY SIMILAR BLOWERS.

The preceding formulas also serve to determine the performance of geometrically similar blowers from the characteristic curves of one blower at a single speed. The character-

istic curves at 1,550 revolutions per minute in Fig. 7 were obtained for a blower having an inlet diameter 17.5 inches. The speed at which a 20-inch blower must operate to give the maximum efficiency, and the corresponding volume discharge and effective static pressure, may be readily calculated by the preceding formulas. There are many problems, however, that would requre rather complicated calculations to solve them. The solution of such problems is easily obtained by means of three additional formulas derived from the preceding ones.

The three formulas are: First, a simple transposition of formula (1), namely,

$$\frac{Q}{D^3} = C_1 N. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Second, a combination of formulas (1) and (2a), eliminating N by dividing Q by the square root of h_w; thus,

$$\frac{Q}{\sqrt{h_w}} = \frac{C_1 D^3 N}{\sqrt{\frac{12 C_2 d_a}{d_w} DN}} = \frac{C_5}{\sqrt{d_a}} D^2. \quad . \quad . \quad . \quad (5)$$

For standard air density $\frac{C_5}{\sqrt{d_a}} = K_1$, or

$$\frac{Q}{\sqrt{h_w}} = K_1 D^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

Formulas (5) and (5a) are simultaneous with formulas (1) to (4), and therefore correspond to the constant efficiency indicated by formula (4). The third formula is a general relation obtained by a combination of the relations (C) and (D) with the fact the efficiency e is equal to the work done. Wh_a divided by the power P required to run the blower; that is,

$$e = \frac{Wh_a}{P} = \frac{60 d_a Q \times h_a}{P} = \frac{60 d_w}{12} \frac{Q h_w}{P}.$$

or,

$$P = \frac{60 \times 62.30}{12} \frac{Q h_w}{e}$$

$$= 311.50 \frac{Q h_w}{e} \text{ foot-pounds per hour, . . (G)}$$

$$= 0.0001\,5732 \frac{Q h_w}{e} \text{ shaft horsepower . (G-1)}$$

The several methods of using these formulas will be illustrated by applying them for determining the performance of any blower geometrically similar to the one having the characteristics in Fig. 7. In a geometrically similar blower every dimension of wheel and casing must bear the same ratio to the corresponding dimension in the blower tested. The blower tested had an inlet diameter of 17.5 inches. Substituting in (1) and (5a), 17.5 inches for D, and the values of Q, N, and h_w from Fig. 7 that correspond to efficiencies of 51.6, 50, 40, 30 and 20 per cent. respectively, we obtain the constants in Table I.

TABLE I.

Efficiency, pr. ct. e	Inlet diameter, inches. D	Rotational speed, rev. per min. N	Volume discharge, cu. feet per min. Q	Effective static pressure, inches water. h_w	C_1	K_1
20	17.5	1,550	3,040	6.51	0.0003660	3.890
30	17.5	1,550	5,050	6.21	0.0006079	6.617
40	17.5	1,550	7,670	5.93	0.0009233	10.285
50	17.5	1,550	11,800	6.12	0.0014205	15.575
51.6	17.5	1,550	14,600	6.31	0.0017576	18.979
50	17.5	1,550	18,000	6.11	0.0021668	23.778

These constants are plotted in Fig. 9 in order to find their values for intermediate efficiencies.

The curves in Fig. 10 and Fig. 11 are plotted from the respective relations:

$$\frac{Q}{D^3} = C_1 N \text{ and } \frac{Q}{\sqrt{h_w}} = K_1 D^2,$$

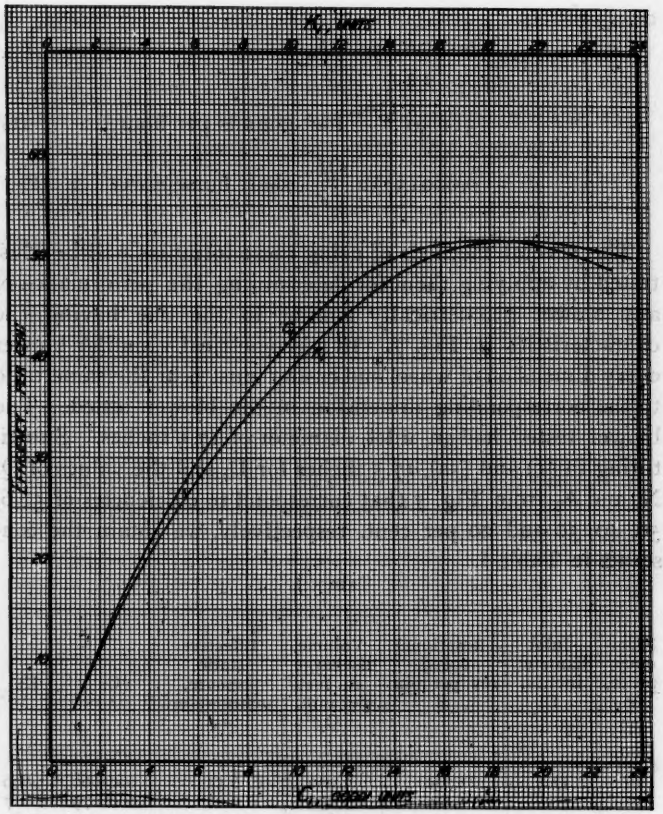


FIG. 9.

for the constants corresponding to the tabulated efficiencies. The curves in Fig. 12 were gotten by cross-plotting from Fig. 11 for the commercial fan sizes noted thereon.

Now, let us solve the problem: What size and speed of blower of the type tested should be selected in order to operate at about maximum efficiency when delivering 30,000 cubic feet of air per minute (at standard density) against a static pressure of 6 inches water? What horsepower is required to drive it? Now,

$$\frac{Q}{\sqrt{h_w}} = \frac{30{,}000}{\sqrt{6}} = 12{,}247,$$

and by calculation or from Fig. 11 we find the diameter should be 25.4 inches at the maximum efficiency of 51.6 per cent. The nearest commercial size (which we will select,) is 25 inches, and it corresponds to an efficiency of 51.6 per cent., as indicated on both Fig. 11 and Fig. 12. Then

FIG. 10.

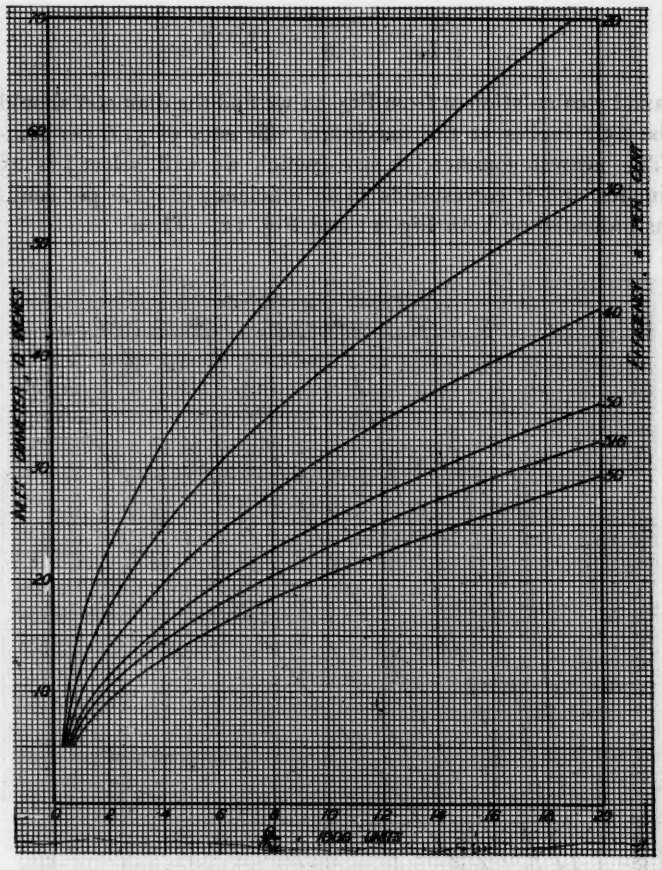


FIG. 11.

$$\frac{Q}{D^3} = 1.92,$$

and we find by calculation (using Fig. 9) or from the curves on Fig. 10 that this ratio at the efficiency somewhat less than 51.6 per cent. requires a rotational speed about 1,070 revolutions per minute. The horsepower to drive the blower will be, from formula (G-1),

$$P = 0.00015732 \frac{30{,}000 \times 6}{0.516} = 54.9.$$

How can we obtain the characteristic curves of this 25-inch double-inlet blower of 1,070 revolutions per minute? The answer appears in Table II. The values in column 4 may be taken from the curves in Fig. 10 or obtained by multiplying

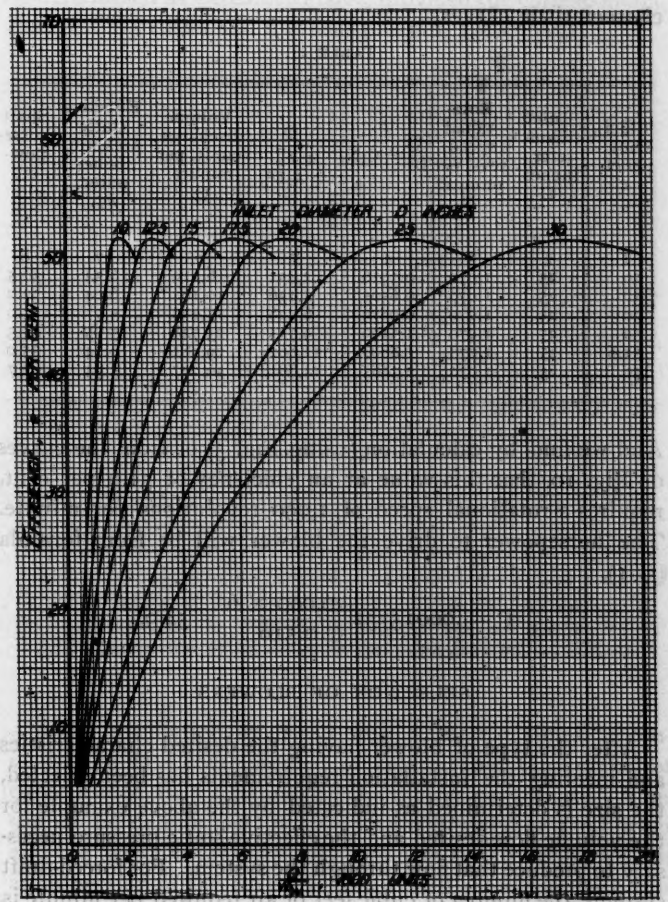


FIG. 12.

the corresponding values of C_1 in Table I by 1,070. Similarly, column 5 may be gotten from either Fig. 11 or Table I. Column 6 results from multiplying column 4 by 15,625 (D^3); and column 7 results from squaring the quotient of column 6 divided by column 5. The shaft horsepower, column 7, is calculated by means of formula (G-1). It is considered sufficient to indicate the method without actually plotting the characteristic curves.

TABLE II.

Efficiency, per ct. e	Inlet diameter, inches. D	Rotational speed, revs. per minute. N	$\frac{Q}{D^3} = C_1N$	$\frac{Q}{\sqrt{h_w}} = k_1D^2$.	Volume discharge, cu. ft. per minute. Q	Effective static pressure, inches water. h_w.	Shaft horsepower. P.
1	2	3	4	5	6	7	8
20	25	1,070	0.3916	2,431	6,119	6.34	30.5
30	25	1,070	0.6505	4,136	10,164	6.04	32.2
40	25	1,070	0.9879	6,428	15,436	5.77	35.0
50	25	1,070	1.5199	9,734	23,748	5.95	44.5
51.6	25	1,070	1.8806	11,862	29,384	6.14	55.0
50	25	1,070	2.3185	14,861	36,227	5.94	67.7

And we find by calculation (using Fig. 5) or from the curves on Fig. 10, that this value at an efficiency of 51.6 per cent. requires a rotational speed of 1,090 revolutions per minute. The horsepower to drive the blower will be from formula (G-1),

$$P = .0001573 \frac{30{,}000 \times 6}{0.516} = 54.9.$$

SELECTION OF BLOWERS.

After the type of blower having the desired characteristics and meeting the installation requirements has been selected, the size is determined as indicated by the above example for the volume flow desired and the effective static pressure necessary to produce that flow through the system. For forced-draft systems, the number of cubic feet of air required per minute is

based on the quantity of fuel to be burned, an allowance being made for leakage. The static pressure necessary in the fireroom is found by experience, the analysis of existing systems giving data for new ones. With a blower selected to meet these conditions at full load, its performance under different conditions due to changes in draft and stack temperature, may be determined as explained above, provided the corresponding changes in air flow and fireroom pressure are known.

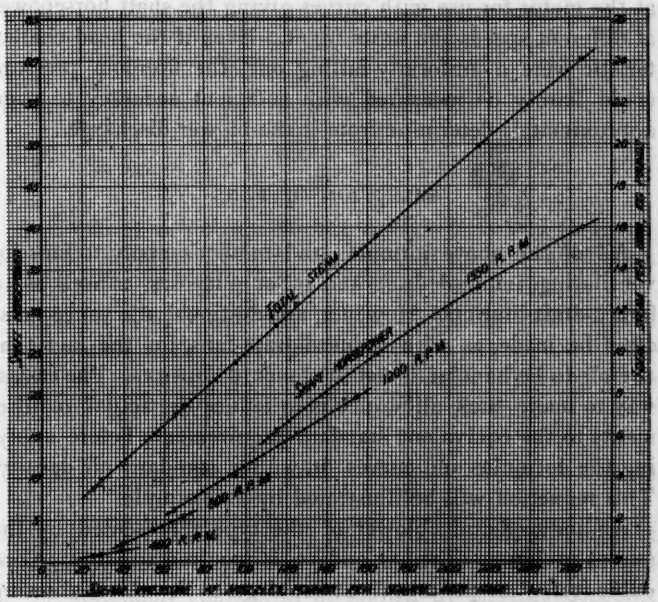


FIG. 13.

After installation on board ship, it may at times be convenient to have some method of determining the number of cubic feet of air discharged by the blower per minute. This cannot be determined by the rotational speed of the blower and the fireroom pressure only, the data usually observed, for an inspection of the curves in Fig. 6 will show that with this blower there may be three different capacities corresponding

to a certain pressure and a given speed. For steam-turbine-driven blowers, the capacity can be determined by noting the steam pressure at the nozzles, curves similar to those in Fig. 13, then giving the shaft horsepower and locating the proper capacity on the curves in Fig. 6. In Fig. 13 is also plotted the curve for steam consumption—a function of the steam pressure only. For motor-driven blowers, the corresponding method would consist in measuring the voltage and current at the motor for use with curves giving the shaft horsepower at different speeds and watts input. Even if the head characteristic were a uniformly falling one, the fireroom pressure could hardly be measured with sufficient accuracy to even approximately determine from it the blower discharge.

SELECTION OF FANS.

All of the above formulas and methods apply to fans for ventilating, induced draft and other purposes, as well as to forced-draft blowers. However, an additional allowance must often be made, as with induced-draft fans, for the fact that the air to be handled is not at standard density. The easiest way to make such allowance is to reduce the actual requirements to the equivalent standard requirements before applying the formulas and curves drawn for standard air density. This reduction will be illustrated by an example.

An induced-draft blower of the type represented by the curves in figures 9, 10, 11 and 12, is required to move 30,000 cubic feet of air per minute at 600 degrees F. After allowing for the stack "draft," it is found that the blower must produce an effective static pressure of 3 inches water between inlet and discharge. What size, speed and horsepower correspond to maximum efficiency? Assuming the densities in the inverse ratio of the absolute temperatures, we have from formulas (1) and (2a) the corresponding values under standard conditions, namely,

$$Q = 30{,}000 \text{ cubic feet per minute;}$$

$$h_w = \frac{460 + 600}{460 + 70} \times 3 = 6 \text{ inches of water.}$$

The problem then becomes the same as the one solved above, giving a 25-inch fan to be run at 1,090 revolutions per minute. The power found for air of standard density was 54.9 horsepower. Under the conditions stated in the problem the power would be from formula (3),

$$P = \frac{460 + 70}{460 + 600} \times 54.9 = 27.4 \text{ horsepower.}$$

Or, the power might be directly found by

$$P = 0.0001\,5732\,\frac{30{,}000 \times 3}{0.516} = 27.4 \text{ horsepower.}$$

As a matter of fact, the curves in Figs. 9, 10 and 11 would not apply accurately to this case, because the test installation was entirely different from that of an induced-draft fan.

It may also be interesting to note that a fan installed in a fixed system, such as certain ventilating systems, always operates at the same efficiency at all speeds. This may be shown by relation (F-1) which is put into the form

$$\frac{Q}{\sqrt{h_w}} = A\sqrt{\frac{d_w}{12kd_a}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (F\text{-}2)$$

The similarity to formula (5), which corresponds to a constant efficiency, proves the truth of the above statement. The truth of the statement could also be shown by plotting constant efficiency curves in Fig. 6 through the points for runs with the same nozzle on the discharge chamber. The corresponding runs at each speed were made with the same nozzle.

SUMMARY.

By considering in detail the energy supplied by a blower or fan to move air or other gas through a system against the resistances encountered therein, there have been derived four

simultaneous equations giving the variation of the volume or weight flow, of the effective static head or pressure, of the power, and of the efficiency with the density of the air moved, and the size and speed of the blower employed. If the variation of the effective static pressure, the power, and the efficiency with the volume flow, is known at any particular air density with a given blower at a constant speed, it is possible to determine by means of these four simultaneous equations what the performance of a geometrically similar blower will be at a different speed and handling air of a different density.

In making such calculations, however, it is best to reduce the test data for a given blower to a standard air density and to make all speed and size conversions for handling air at the standard density. In selecting the proper size of blower or fan for a particular installation, the actual requirements should be reduced to these corresponding to the standard air density before attempting to make the selection. By this procedure, it becomes possible to plot groups of curves, which will enable the selection to be readily made.

The test measurements for these curves should correspond as nearly as possible to the actual installation conditions.

PROPULSIVE MACHINERY FOR SUBMARINES.

By Mr. G. Berling.

Translated from the German "Transactions of the Society of Naval Architects," 1913, by Lieut. W. P. Beehler, U. S. N.

INSTALLATION WITH STORAGE BATTERIES, ELECTRIC MOTORS AND DIESELS.

To date practically all submerged propulsion of submarines is accomplished by means of storage batteries and electric motors. For surface propulsion small "Sousmarins" are fitted for electric drive, while the larger boats are generally fitted with oil engines or, in some few cases, with steam engines.

Boats fitted with oil engines for surface propulsion, with lead batteries for submerged running, will be designated as the "oil-electric boats."

Those having a steam installation for the awash condition and which are fitted with storage batteries for submerged running will be referred to as the "steam-electric boats."

Reversible 2-cycle and 4-cycle oil engines of considerable size have been developed in recent years and have proved practicable in many tests. The whole machinery installation, including the air flasks for all purposes, auxiliary machinery, shafting, propellers, etc., weighs about 28–32 kg. (60–70 pounds) per H.P., while the main engines alone come to about 24 kg. (53 pounds) per H.P. These engines will require about 190–220 g. (.42–.485) pounds of fuel per H.P. hour.

In recent years the lead storage batteries have been developed to a point where they are both reliable and durable, and one may expect a life from them of about 400 charges. This means for the average submarine making practice runs a life of from 5 to 6 years. In the best modern submarines the stor-

age batteries with a 3½-hour discharge rate weigh about 35.5 kg. (78.3 pounds) per H.P. hour delivered.

The iron-nickel cells of Edison have not yet been tried out, but are in the process of construction.

The development of the storage battery, electric motors and control apparatus was already far advanced before they were utilized for submarine propulsion.

The electric motors, including all wiring and all control apparatus, weighs about 60 kg. (132.5 pounds) per H.P. at a 3½-hour discharge rate.

The sum total of these weights comes to 30 plus 124 plus 60, or 214 kg. (472 pounds) per H.P. hour for submarines. A modern large torpedo boat has an equivalent weight of 18 kg. (39.7 pounds) per H.P. for the propulsive machinery. From the above it is evident that the combined weight per H.P. of propulsive machinery for submerged and surface drive of a submarine is about eleven times as great as for a modern torpedo-boat.

While with torpedo boats about 48–50 per cent. of the total building weight may be allowed for the machinery, yet with submarines only about 28–32 per cent. may be applied to the machinery installation. Taking into consideration the fact that for the same size hull a submarine must have 20 per cent. greater weight than the torpedo boat, on account of the additional pressures, this 20 per cent must be subtracted from the allowed machinery weights. On account of the heavier installation for submerged running it is evident that of this reduced machinery weight only one-eleventh may be as usefully applied as in torpedo boats. It is clear, therefore, why it is impossible to obtain greater power and speed with such machinery installations.

In his work "Bateau Sous Marins a grande vitesse sous l'eau" C. del Proposto derived the following formula for speed from the British Admiralty formula, viz:

$$v = \left(\frac{aC}{A}\right)^{\frac{1}{3}} \times D^{\frac{1}{6}},$$

where
a = Ratio of machinery weights to displacement;
A = Machinery weight per H.P.;
C = Constant;
D = Displacement;
v = Speed in knots.

Since D is under the ninth root while A stands in the denominator under the cube root, it is evident from the formula that an increase in the total machinery weight and corresponding displacement will have little effect upon the speed unless the value of A is considerably reduced. That means that with the heavy electric machinery installation no greater speed is attainable, for either submerged or surface propulsion, with moderate displacements.

It follows, therefore, that any increase in the surface propulsion machinery must necessarily cause a decrease in the underwater speed and vice versa.

The following comparative table has been computed in which in Part I, the weight of the electric installation for submerged running has been taken as constant at 126.5 tons, while for Part II the weight of the electric installation has been increased in the same proportion as the total machinery weight was increased.

The tank-model towing data and the propulsive coefficients measured on several boats have been applied, and the remaining data has been computed from the British Admiralty Formula for Similar Ships (Laws of Comparison).

From this it is evident that the oil-electric boats of moderate size have reached the upper limit of propulsive effect.

In particular, a one-sided increase of the power and speed for surface drive will result in very poor propeller efficiencies for submerged running, and the speed given under Table I for the higher displacements must be reduced for submerged running on account of the poor propeller efficiencies resulting from propellers designed to give high propulsive coefficients at the higher surface speeds. It is of the greatest importance that

the relation between the surface and submerged speeds be clearly understood in order that the requirements may be met.

When a submarine running submerged exhausts its battery capacity it must come to the surface to re-charge its storage batteries. From this is clearly seen the necessity for large

TABLE I.

	Lead.	Edison.
1. Composition of elements	Lead plates in sulphuric acid.	Iron-nickel in potash solution.
2. Forcing at a 1-hour discharge rate.	Permissible	Inadmissible.
3. Cleaning of cells	Every 2 years	Every 2 years.
4. Weight per H.P. × hrs. at 3½-hr. discharge rate.	35.5 kg., 78.5 lbs.	32.5 kg., 76.5 lbs.
5. Space in cubic dm. per H.P. hrs.	11.5 c. dm., 688.0 cub. ins.	14.5 c. dm., 784.0 cub. ins.
6. Cost per H.P. hour	80.00 M., $20.00	285.00 M., $73.25.
7. Mean voltage drop between cells.	1.92 volts	1.2 volts.
8. Working rate in amp. and hours.	0.9	0.75.
9. Working capacity in watts and hours.	0.75	0.55.
10. Voltage drop on discharge	10 per cent	18 per cent.
11. Normal life	400 cycles (experiment).	800 cycles guaranteed.

COMPARATIVE TABLE OF OIL-ELECTRIC SUBMARINES.

PART I.

Displacement, in tons.	610	630	655	710	865	1,095
Machinery weights, in tons (surface drive)	48	54	61	76.5	121	186.5
Machinery weights, in tons (submerged drive)	126.5	126.5	126.5	126.5	126.5	126.5
S.H.P. (surface)	1,512	1,700	1,920	2,410	3,815	5,890
S.H.P. (submerged), 1-hour duration	1,100	1,100	1,100	1,100	1,100	1,100
Surface speed, in knots.	16	16.5	17	18	20	22
Submerged speed in knots, 1-hour duration	10.1	10	9.9	9.72	9.3	8.8
Submerged speed, in knots, 3½-hrs. duration	8.08	8	7.93	7.8	7.44	7.05

PART II.

Displacement, tons	450	630	825	1,400	3,500
Machinery weight, surface, tons	38.5	54	71	120	300
Machinery weight, submerged, tons	90	126.5	165	280	700
S.H.P. surface	1,215	1,700	2,235	3,780	9,450
S.H.P. submerged, 1 hour	782	1,100	1,435	2,335	6,100
Speed, in knots, surface	16	16.5	17	18	20
Speed submerged	9.7	10	10.3	10.9	12.1
Speed submerged, 3½ hours	7.8	8	8.2	8.7	9.7

charging dynamos and an accumulator system capable of carrying large charging currents. Herein lies the important criterion of the oil-electric installations.

II.—STEAM ENGINE INSTALLATIONS.

The inadequacy and unreliability of the early internal-combustion engines, storage batteries and electric motors led the French to experiment with steam installations as early as the year 1890.

While the internal-combustion engines require extensive cooling with sea water, and therefore radiate but little heat, and the electric motors are readily cooled with water circulated in small pipes, yet with steam installations it has been found extremely difficult to maintain bearable temperatures in the inclosed living spaces, particularly in rough weather when the vessel is battened down. The principle of the steam engine requires the avoidance of all condensation losses and the insulation of heat; yet with thickest silica and felt clothing it is impossible to carry off all the heat radiated, and which therefore accumulates in a crowded submarine where skylights are necessarily impossible. For this reason all bulkheads not cooled by the outside water must be built in two watertight thicknesses and cooled by water forced through cooling pipes between the bulkheads. This cooling water must be kept constantly circulating through the pipes by special pumps.

Internal-combustion engines assist in the ventilation of the living spaces by their own use of air; but with steam installa-

tions, on the other hand, large ventilators must be provided in order to supply sufficient air and to further assist the draft of the small smoke pipe, as the latter must be made both watertight and removable, or capable of being closed up.

As a fuel for the boilers, oil is the only possibility, as coal, on account of the difficulty of trimming and weight, would present unsurmountable obstacles.

The boilers must be built in such a manner as to leave free passage of the personnel past them, and take up nearly the whole 'thwartship space, even though specially constructed with part of the breadth cut down. While the Diesel engine requires about 220 g. (.485 pounds) of oil per H.P., the steam-engine installations require about 3.5 times this amount. This makes it almost impossible to obtain more than 15 knots surface speed without considerably reducing the radius of action.

In spite of these enormous difficulties, Labeuf built his first submarine, the *Narval,* in 1879, with steam engines for surface propulsion and electric drive for submerged running. The lack of good, safe oil engines caused quite a few of these steam-electric submarines to be built in France.

This type of machinery for submarine propulsion must pass into ancient history together with the oil-electric installations. Since the steam engine occupies more space than an oil engine of the same power the other spaces of the boat must be more crowded, especially the living spaces; but, in any event, of the two types the steam installation will have the smaller radius of action, other things being equal.

III.—STEAM SODA-BOILER INSTALLATIONS.

Of much greater importance is the steam installation for submarine boats fitted with a "soda" boiler for supplying the engine when running submerged. These soda-boiler installations are the idea of the German inventor, Moritz Honigmann, of Grerenburg, near Aachen (Aix-la-Chappelle).

For this purpose the following appliances are necessary, viz:

1. One boiler (water-tube) fitted for oil burning for surface propulsion.

2. One steam-engine installation with auxiliaries, pumps, condensers, etc.

3. One warm-water container that can be stowed in sections.

4. One double boiler (CD), consisting of a container for a solution of sodium hydroxide (C), surrounded by a vessel containing water for the formation of steam (without the necessity of burning fuel).

When cruising on the surface the boilers fitted for oil burning are in use and the steam is generated in the ordinary manner.

When the boat is running submerged the vapor from the low-pressure cylinder is led to container for the sodium-hydroxide (C) and heats the solution. The vapor is readily taken up by the solution and condensed, giving up its latent heat of vaporization to the sodium-hydroxide. In addition to this the soda is further heated by going into solution as the steam is condensed, thereby supplying enough heat to boil the water in the surrounding flask and generating sufficient steam to supply the main engines. The water of condensation from the condenser is forced into the tubes of the soda boiler (D) together with sufficient make-up feed from the warm-water container to keep the water level constant in the drum. When this method of steaming is commenced only a slight pressure exists in the container "CC," as the concentrated solution absorbs the vapor readily. The more the solution strength is weakened and heated up the less will be the absorbing power of the solution and the greater will be the back pressure, until finally the steam for heating the solution must be taken from the I.P. cylinder of the main engines.

At the conclusion of the submerged run no further steam is led to the container "C," as that is already full, and now serves only as a heater to vaporize the water in "D." The balance of the steam in "D" is drawn off in the engine, condensed and forced into the feed heater "B." This ends the submerged run.

Before a new submerged run can be undertaken the solution of sodium hydroxide must be vaporized by starting the boiler "A" under oil, passing some of the steam thus generated in the tubes of the soda boiler " D," thus heating and vaporizing the solution. The water then condensed in " D" is used as make-up feed for the boilers "A." The steam from the hydroxide solution is led to the condensers, condensed and pumped to the warm-water reservoir. When the warm-water container is nearly filled the boiler " D" is then pumped up to normal water level. Finally, the boilers "A" and " D" are filled to their normal level, " B" is full and "C" contains a concentrated solution of sodium-hydroxide.

This installation must be so arranged that there will be no change of trim resulting when running submerged. For this reason two soda boilers are fitted with a warm-water container situated between them. Before beginning the submerged run the water in " B" must be brought up to a high temperature by means of live steam taken from boiler "A," in order that conditions in " D" will be favorable to the formation of steam.

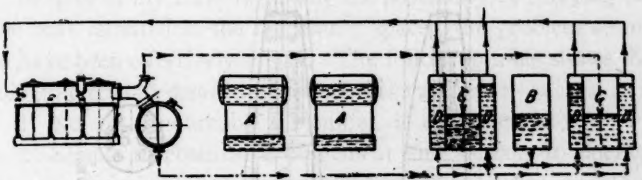
When running awash and vaporizing the solution simultaneously, Figures 1 and 5 are combined, so that part of the steam goes to the engines while the remainder is utilized in vaporizing the solution, the water of condensation being sent to both the boiler "A" and the container " B."

Figure 6 shows a proposed submarine of 700 tons fitted with soda boilers. Amidships is the central-control station, while abaft this is the oil-burning boiler room, machinery space and the condensers. Forward are the two soda boilers with a warm-water container between them. Forward and abaft these are the living spaces and the torpedo rooms. The living spaces consist of securely built-in rooms, to which much thought must be given if comfortable boats are to be built. The plan was developed by the Weser Construction and Shipbuilding Co., of Bremen. This company contemplates the construction of such a boat after having made exhaustive and comprehensive experiments and tests with steam-soda installations.

STEAM-SODA INSTALLATION.

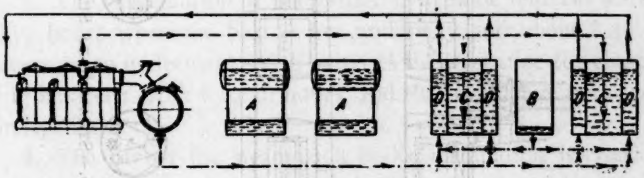
SURFACE RUN.

FIG. 1.



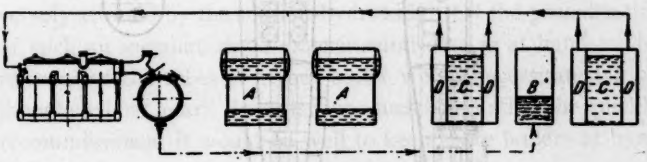
SUBMERGED RUN (a)

FIG. 2.



SUBMERGED RUN (b)

FIG. 3.



SUBMERGED RUN (c)

FIG. 4.

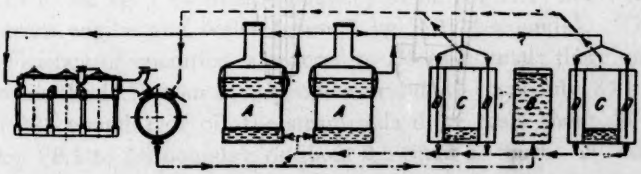


FIG. 5.—BOILING OUT SOLUTION ON SURFACE RUN.

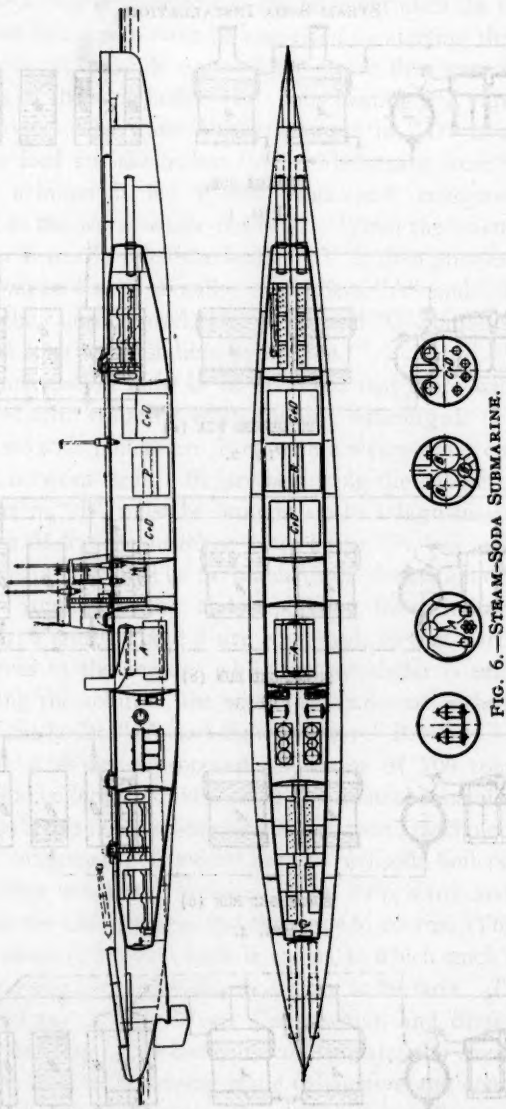


FIG. 6.—STEAM-SODA SUBMARINE.

As regards the practical and theoretical comparisons of the soda boilers with other installations, no details have reached the public since the publication of the "Journal of the Society of German Engineers" in 1884 and 1885. Figures on the power, weight and space of such boiler installations can only be given with certain reservations. I, myself, can only regard them as probable.

In spite of my fears regarding the possibility of carrying off the heat radiated in the machinery spaces, the problem seems to have been effectively solved. The following table shows the advantages and disadvantages of this type, viz:

1. No unsurmountable difficulties of construction.

2. Simple mechanical arrangement and as easy to operate as any steam installation; few repairs, thus saving work for the crew.

3. The vaporization of the sodium-hydroxide requires about 1½ hours when the boat is not underway and about 3 to 4 hours when underway. With batteries the least time for charging has been from 6 to 10 hours, and the boats have been less independent.

4. The life of the steam-soda boiler installation is considerably reduced by the action of the soda on the steel. There are materials, however, such as nickel-steel, that are not adversely affected by the sodium-hydroxide. Of the probable life of such an installation no experimental data is at hand. The renewal of the tubes of either boiler would necessitate taking the whole boat apart. In order that such boats might be quickly recommissioned it would be well to keep spare boilers at hand in reserve.

5. Radius of action.—While the oil engine requires from .19 to .22 kg. (.42 to .485 pounds) of oil per H.P. hour, the steam engine and boiler takes .7 kg. (.1.54 pounds). The efficiency of operation is reduced to .73 of normal; therefore, while the H.P. hour for the oil-electric drive costs only .35 kg. (.723 pounds) of oil, the steam-soda drive costs about .96-1 kg. (2.1 to 2.2 pounds), or about 2.8 times as much. A boat,

therefore, carrying about 90 tons of fuel oil and having a soda-boiler capacity of about 3,000 H.P. hours should be able to cruise independently in enemy waters for about one week and to renew its capacity for submerged running about 5 times. The latter operation costs about 15 tons of fuel oil. There remains about 75 tons of fuel oil for surface drive with the boat illustrated in Figure 22, and it should be able to cover 1,600 miles at 13 knots. The above is always a fairly good achievement.

6. The surface and submerged installations are capable of forcing in the same manner as torpedo boats, and the boat in Figure 6 should be capable of 20 knots on the surface and 12½ knots under water for from 1¼ to 1½ hours.

7. The submerged radius of action appears to be somewhat greater than that of the oil-electric boats for short runs.

8. The weight requirements are smaller for this installation, in that the surface installation weighs only from 17.3 to 20 kg. (38 to 44 pounds) per H.P., and no known machines can approach the output for the submerged installation in weight per H.P. The capacity for submerged running is given by some tests (which must be checked by further experiments), as 38 kg. per I.H.P. per hour (84.6 pounds), while other tests give 48 kg. (106 pounds). For a 3½-hour run the figures are 135 to 168 kg. per I.H.P. hour (300 to 370 pounds). The combined weight per H.P. is therefore only from 150 to 188 kg. (330 to 413 pounds) as opposed to 214 kg. (473 pounds) for the oil-electric installation.

9. The space requirements for this installation are comparatively high. For this reason the allowed weights of machinery cannot always be utilized on account of lack of space, and, in addition, the boats must be ballasted. The latter is necessary on account of the high center of gravity of the machinery and the necessity for obtaining stability in the boat.

10. Finally, the dangers of accidents and breakdown, aside from the corrosion due to the soda, are not more probable than with any steam installation.

However, the steam-soda drive will reach its upper limit as has the oil-electric installation. At all events I have developed the following table from the most reliable data obtainable. The soda boiler in this table is set at a constant weight of 115 tons, and only the steam engine and oil-burning boiler weights are increased with increased displacement.

In this table for surface cruising the coefficient

$$\frac{\text{surface S.H.P.} \times \text{slip H.P.}}{\text{surface I.H.P. (engine)}} = .44$$

$$\frac{\text{submerged I.H.P.}}{\text{surface I.H.P.}} = \frac{1}{2}$$

$$\frac{\text{submerged S.H.P.}}{\text{submerged I.H.P.}} = 0.35$$

$$\frac{\text{total machinery weight}}{\text{displacement (surface)}} = 28.1 \text{ per cent.}$$

Steam-Soda Installations.

Displacement.	523	550	580	665	791	975	1,265	1,475
Weight soda boiler...	114.6	114.6	114.6	114.6	114.6	114.6	114.6	114
Weight steam mach. and 3 El. machine.	34	42	51	75	111	163	246	306
E.H.P. surface.....	1,723	2,165	2,670	4,000	6,000	8,890	13,500	16,800
E.H.P. submerged ...	861	1,082	1,335	2,000	3,000	4,445	6,750	8,400
Surface speed	16	17	18	20	22	24	26	27
Submerged speed.....	10	10.6	11.2	12.5	13.7	15	16.2	16.9
Submerged radius of action, miles ..... At the above speeds.	35	29.4	25.2	18.8	13.7	10		

INSTALLATION WITH COMPRESSED AIR IN NICKEL-STEEL FLASKS WITH OIL ENGINES FOR SUBMERRGED PROPULSION.

The striving after increased output and greater speed led the engineer Del Proposto, in 1909, to work out the scheme, Figure 7, in which oil engines are used as the propulsive machinery for submerged as well as surface running. Atmospheric air is taken in through the compressor, when running

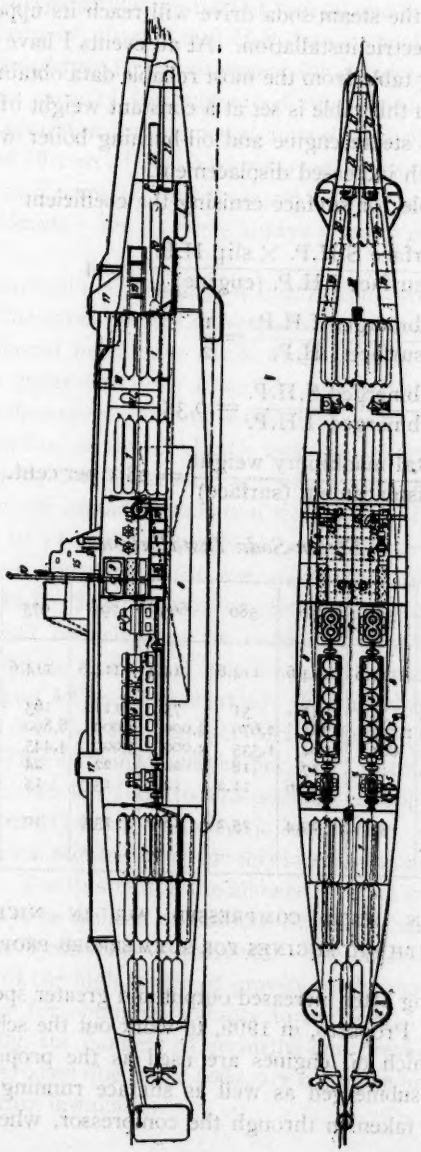


FIG. 7.—SUBMARINE WITH GREAT SUBMERGED SPEED.

awash, in high-tensile light nickel-steel flasks wound with steel wire and compressed to 250 atmospheres. When running submerged the air in these must be expanded in the compressors to about one atmosphere, thereby giving up the energy resulting from the high pressure to the crankshaft of the compressors. The compressors are therefore warmed by the hot circulating water from the oil engines in order to avoid the low temperatures (due to expansion). The expanded air is then drawn into the machine and used in the combustion of the fuel oil in the working cylinder. The exhaust gases are led through condensing flasks, where the steam is condensed, giving up its latent heat to the circulating water. The balance is forced overboard by means of pumps. It is very probable that such an installation can be made practicable for submarine propulsion.

In the proposed scheme, however, all machinery parts are subjected to very high stresses and have very light weights without a sufficiently large factor of safety; for instance. 34 4/10 tons for an installation of 1,700 H.P. rating, that is, 18 kg. per I.H.P. for the complete oil-engine installation, including intake and outlet flasks, all piping, bearings, propellers, etc. Oil engines including all of the above mentioned parts, according to results given by carefully tested engines, must weigh 58 tons, or have a weight of 31 8/10 kg. per I.H.P. The four compressors with large condensing apparatus, which are placed at 27 6/10 tons, would therefore come to about 49 tons. With such small weight requirements there could then be developed a submarine with only 435 tons displacement capable of 1,400 sea miles awash at 16.45 knots speed, and submerged of 27.6 miles at 16.6 knots speed. This scheme possesses still further advantages, for instance, by simply driving the air motors of the compressor without heating by the exhausts. Such a drive could be accomplished for from five to ten hours, provided, of course, the machine was not put out of commission by freezing.

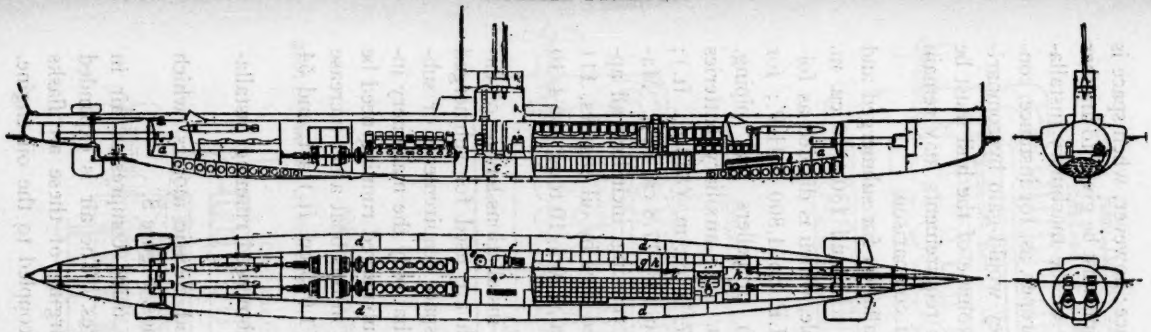
This scheme has only been tried out on two small boats and

two reserve torpedo boats, and is valuable for its high submerged speed for submarines. The machinery space contains two 4-cycle Diesel motors of 6 cylinders each, and in the hold 4 compressors. The space aft and the space forward of the hood are crowded with air flasks. Very crowded and uncomfortable living spaces are located under the hood in the forward torpedo room and between the air flasks. At the bottom of the hood is an electric battery.

COMPARISON OF MACHINERY INSTALLATION FOR SUBMERGED RUNNING, AND THEIR OUTPUTS.

In order to arrive at any practical results from these statements one should not take any particular scheme which happens to show to good advantage, but should compare all installations for submerged running on some common basis, in order that the proper conclusions may be drawn as to their value. While I may not always be in a position to make these comparisons with installations which have been successfully tried out, still, in order to make a practical analysis as a basis for future computations, I hope that all theories which I now advance may be corroborated as soon as possible by careful experiments, in order that a clear insight may be obtained regarding the relative values of the individual modes of propulsion already devised and which may shortly be completed. In this connection a rational boat construction will be analyzed consisting of sufficient armament, convenient living spaces and diving rudders, etc. This boat will be compared successively with various types of machinery installations without any other change of detail. In this manner all weight and space considerations will be kept correct. The various machinery installations will therefore be put in a space sufficient to contain them, and also within the allowed weight. Regarding Figure 8 it is evident that the following conditions must hold: machinery space must extend from the after engine bulkhead to the conning tower with the accumulator space forward. It may happen in some cases that it would suit the installation better

Germania Submarine



a Trimming tank
b Torpedo tank
c Water ballast
d Bunker
e Main bilge pump
f High pressure compressor motor
g Locker
h Water closet.
i Commanding Officer
k Officers
l Deck officers

FIG. 8.

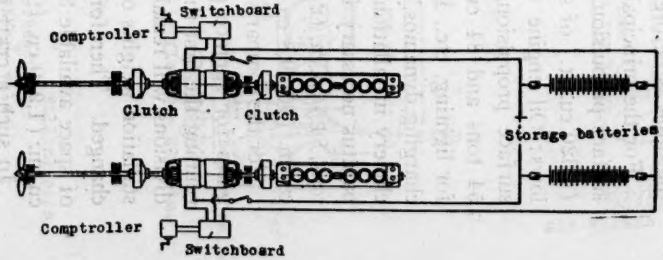


FIG. 8.

to have a different allotment of space; however, when space is taken from one part the same amount must be given to another part. In case, however, some part of the machinery installation is situated outside of the hull proper, as, for instance, condensers or compressor air flask, they would gain by comparison, and for this reason the total volume of the hull must be diminished in order that the space requirements may remain the same as before, for purposes of comparison.

For the principal machinery installation for submerged and surface propulsion about 180 5/10 tons and 180 4/10 cu. m. (6,320 cu. ft.) of space are available, which is divided as follows: Oil engine from 1,700 S.H.P. to 1,800 S.H.P.; for surface propulsion, complete with propellers and piping, 54 tons and 84 cu. m. (2,930 cu. ft.); auxiliary batteries for lighting, etc., 12 5/10 tons, 9 7/10 cu. m. (343 cu. ft.); charging dynamos, 3 tons, 2 2/10 cu. m. (77.8 cu. ft.). Machinery installation for submerged running, including all apparatus necessary for the submerged dive by oil engines, 111 tons, 84.5 cu. m. (2,990 cu. ft.); total, 180 5/10 tons, 180 4/10 cu. m. (6,350.8 cu. ft.).

For the comparison of machinery installations having oil engines for surface propulsion which are used for submerged running the first three weight and space requirements or subdivisions will remain the same, so that only the machinery installation weights of 111 tons for submerged running need be changed. Therefore, there remains in this boat a net increase of space available 30 5/10 cu. m. (1,075 cu. ft.) aft, and 54 cu. m. (1,910 cu. ft.) forward.

In such a craft the following submerged running installations are proposed:

1st. Electric storage batteries and electric motors, which may also be used as charging dynamos, Figure 8.

2d. The first air plan (Fig. 9), using compressed air in nickel-steel flasks at 200 atmospheres. The air is expanded without doing work. For the charging of these air flasks compressors are installed, directly coupled to the oil engine,

MACHINE INSTALLATION, COMPRESSED AIR SUBMARINE, I

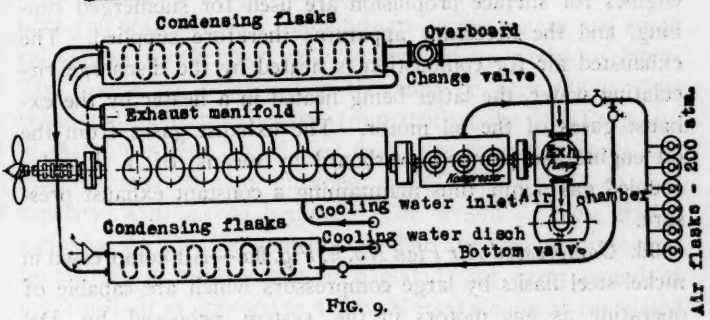


FIG. 9.

MACHINE INSTALLATION, COMPRESSED AIR SUBMARINE, II.

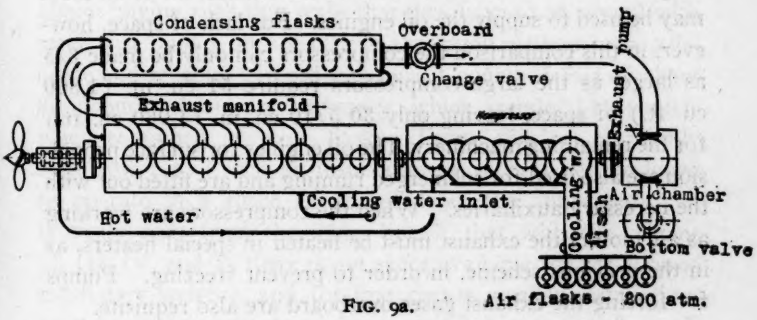


FIG. 9a.

MACHINERY INSTALLATION USING OXYGEN.

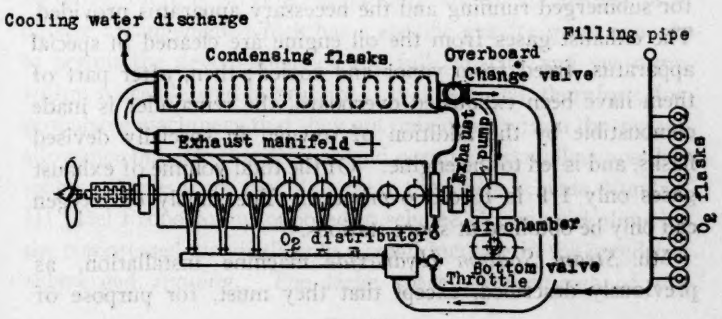


FIG. 10.

capable of completely charging them in 5 hours. The oil engines for surface propulsion are used for submerged running, and the necessary apparatus therefore supplied. The exhausted air for combustion is heated by the discharge circulating water, the latter being heated in a heater by the exhaust gases of the oil motor. The exhaust gases from the oil engine are pumped overboard by means of a loosely-coupled gas pump, thus maintaining a constant exhaust pressure.

3d. *Compressed Air Plan No. 2, Fig. 9a.*—Air compressed in nickel-steel flasks by large compressors which are capable of operating as gas motors in the system proposed by Del Proposto. According to this scheme the compressors are so large that while in use as air motors the exhaust air from them may be used to supply the oil engines. For lack of space, however, in this comparison the compressors can only be made 3/5 as large, as the large compressors require 51 cu. m. (1,800 cu. ft.) of space, leaving only 30 5/10 cu. m. (1,060 cu. ft.) for the remaining machinery. The oil engines for surface propulsion are also used for submerged running and are fitted out with the necessary auxiliaries. When the compressors are working as air motors the exhaust must be heated in special heaters, as in the previous scheme, in order to prevent freezing. Pumps for forcing the exhaust gases overboard are also requisite.

4th. Figure 10. *Compressed Oxygen and Nickel-Steel Flasks.*—The oil engines for surface propulsion are also used for submerged running and the necessary apparatus provided. The exhaust gases from the oil engine are cleaned in special apparatus, freed from vapor and cooled; then, after part of them have been exhausted overboard, the remainder is made combustible by the addition of oxygen in specially devised flasks, and is led to the engine. Of the total volume of exhaust gases only 1/9 is forced overboard. The supply of oxygen can only be obtained at shore stations.

5th. *Steam Sodium Hydroxide* machine installation, as previously described, except that they must, for purpose of

comparison, be somewhat cut down in their output in order to come within the weight and space requirements. I will not bore you with tiresome computations on the weight and space calculation which could only be checked by weeks of labor. The results are contained in the table on page 0000. The calculations differ somewhat from that of the Del Proposto, in that 6 per cent. nickel-steel flasks are computed for a 450 mm. diameter, 12 mm. thickness and 300 liters (10.8 cu. ft.) capacity, with special bottoms. The weight of these flasks, inclusive of the air contents at 200 atmospheres pressure, necessitates a total weight of 6 pounds for every pound of contained air as opposed to 4 pounds total weight as calculated by Del Proposto. I believe it to be dangerous to the personnel to have air flasks containing compressed air at 250 atmospheres having such thin walls that they have to be wound with steel wire, and I do not believe the latter method to be adequate and reliable. In order to burn 1 pound of fuel oil about 24 pounds of air are necessary, which weighs 144 pounds when contained in steel flasks. From this it is readily seen how heavy such an installation must be when designed to operate in a space closed off from the atmosphere. It is easily seen from the table, line 7, that with the scheme II, III, IV, and V, there is not space available for the allowed machinery weights, and for that reason smaller installations must be used and the remaining space filled up with ballast. It should be noted that, aside from the very heavy lead storage batteries, there is no other commonly known submerged propulsive arrangement which has a specific gravity great enough to permit the development of the full capacity and yet come within space allowed in practice. It is essential, therefore, for propulsive machinery that they not only come within the weight allowance but also that they should occupy the smallest possible space. Aside from that it is clear that with scheme Number III (Del Proposto) as opposed to scheme II, the work done by the compressed air yields only a small increase in the speed for submerged running. The large space requirements of the

compressors causes a loss of capacity and radius of action which is not counterbalanced by the small gain in speed in boats of the same displacement.

In columns I, II, III and V the combined capacity is not great enough to drive the charging machines. In column IV no charging is necessary, for technical reasons, thus giving it an advantage in this case.

In my experience I have found it impracticable to compare schemes which have been tried and proven with those which are as yet undeveloped, as with columns I and III. The previously deduced results for the steam-soda installations have been carried out in this table.

The installation given in column IV submerged drive, using Diesel's on oxygen in nickel-steel flasks, was tested out on a 30-H.P. engine at the works of the Germania Co., and functioned well on the test stand. It proved to be a source of danger to the engine and boat, however, when, as actually did once occur, the oxygen-separating apparatus was not properly regulated. After missing once it supplied too large a quantity of oxygen to the working cylinder and excessive pressures were developed in the engine. Aside from that, it is questionable if the supply of oxygen at the naval stations could be made sufficient for military purposes. It follows, therefore, that row 17 must yield a smaller output (1,700 H.P.) for this reason, viz:

In the first place the back pressure on the exhaust necessitates reckoning a loss of 12½ per cent. of the power developed normally, and in the second place the resistance of the hull is greater when running submerged. As the same propeller is used for the surface and submerged drive, the propulsive coefficient changes with change in speed when running submerged and results in a further loss. The latter factor means a power loss of 25 per cent., making a total loss of power of 34 per cent. In the steam-soda boiler installations, however, a loss of about 50 per cent. must be allowed for in order to compensate for the increase in the back pressure in the soda container when running submerged.

The following curves show the manner in which the weight and space requirements increase with demand for increased capacity and output for the different systems.

For submerged-drive machinery installations it is necessary to distinguish whether or not the weight and space are to be utilized for obtaining power or capacity (K).

Calling G_l the weight depending upon the H.P. developed;
G_k the weight depending upon the capacity (K) in H.P. hours;
L the power developed in H.P.;
the unit weight per power developed (a_1) is given by:

$$a_1 = G_l/L \text{ in kg. per H.P.;}$$

and for weight depending on capacity

$$b_1 = G_k/K \text{ in kg. per H.P. hours.}$$

These weights are calculated for the previously considered installations and are included in the table. When one may assume that for a given displacement the previously calculated values of a_1 and b_1 continue to hold, then for a scheme requiring the development of "x" H.P. and "y" H. P. hours capacity, the combined weight (P) is given by the formula:

$P = a_1x$ plus b_1y in kg. units.

The same thing holds good for the space requirements for the submerged-propulsion installations, and the following analagous formula may be developed:

$$R = a_2x \text{ plus } b_2y \text{ in liters,}$$

Where $a_2 = b_{l/L}$ in liters per H.P.
$b_2 = b_K/K$ in liters per H.P hour.

In this formula R represents the space requirements in litres.

The values of a and b are given in lines 13 and 11 of the table. From the above data the curves in Figs. 28 and 29 have been plotted, showing the increase in weight and space requirements (without ballast), with increase of ca-

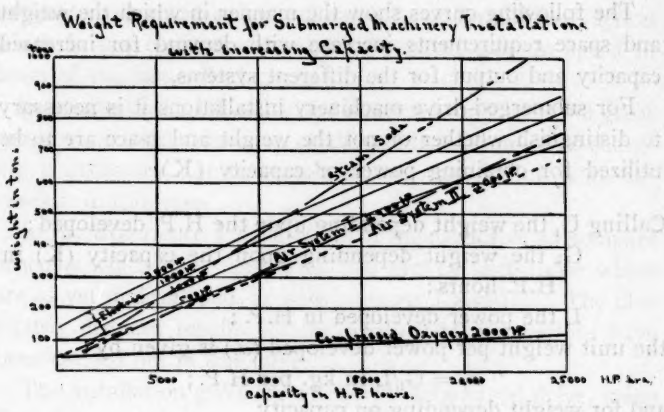


FIG. 28.

pacity (without re-charging). The curve showing the electric installation must be corrected on account of the fact that for the same power developed, the capacity increases with an increase in the discharge rate y/x; and with a slower discharge rate the weight and space requirements decrease. As soon as the limit of space allowed has been reached the practicable

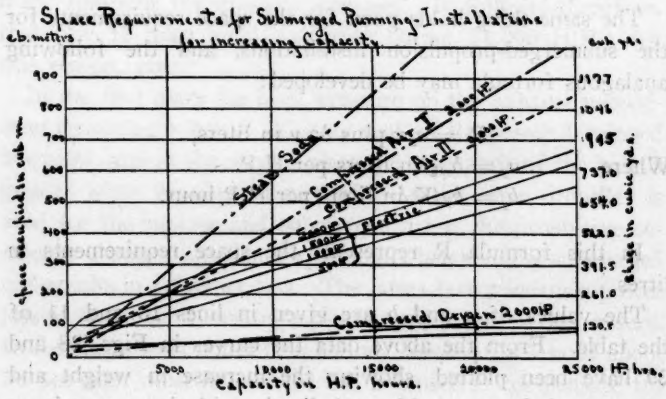
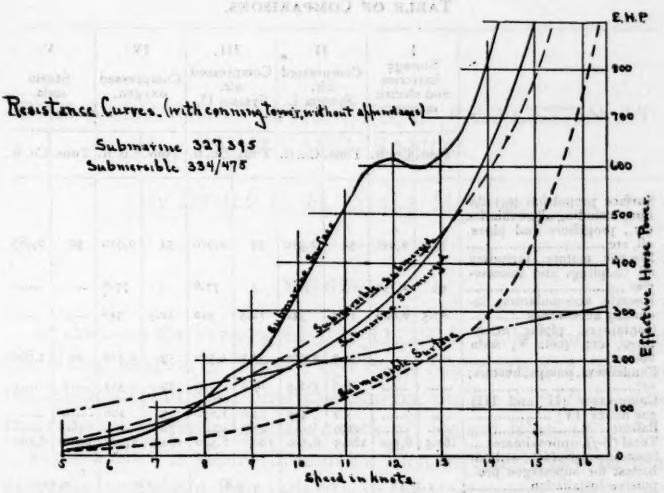


FIG. 29.



capacity must be determined from the curves 29, while the weight allowance for this capacity is picked off the curve 28. The following sequence will be observed in the different schemes for increased weight and space requirements with increased capacity, viz: Soda boilers, air I, air II, oxygen; the latter being the most favorable.

This limit will always be low in comparison with other men-of-war as long as the propulsive requirements of submarines necessitate the employment of two fundamentally difference types of machinery. The failures have been due, as has been previously shown, 1st, to the weight of the hull, and, 2d, to the fact that submarines with single motors for both submerged running and surface cruising must carry the necessary oxygen in flasks, which considerably increases the weight and space required.

Although the present-day submarines show a high state of development of the shipbuilding art, they have not as yet reached the requirements demanded by the Admiralty, and in the future development of these boats there is a wide field of interesting labor for the naval architect.

TABLE OF COMPARISONS.

.....	I Storage batteries and electric motors.		II Compressed air. System I.		III Compressed air. System II.		IV Compressed oxygen.		V Steam soda.	
	Tons.	Cu. ft.	Tons.	Cu. ft.	Tons.	Cu. ft.	Tons.	Cu. ft.	Tons.	Cu. ft.
1. Surface propulsion installation, including all auxiliaries, etc., propellers and pipes, oil, etc	54	2,910	54	2,910	54	2,910	54	2,910	50	2,485
2. Electric motors, including all couplings and accessories	43	1,152	3	77.6	3	77.6	3	77.6	.....	.....
3. Electric accumulators, including accessories	83.5	2,242	12.5	342	12.5	342	12.5	342	.....	.....
4. Containers, piping armatures, etc. (col. V, soda boilers)	.....	.....	54.3	2,362	35.5	1,552	59	2,510	82	3,880
5. Condensers, pumps, heaters, etc	.....	.....	8.6	116.5	19.0	353	19	353	.....	.....
6. Compressor (II and III) gas mixer (IV)	.....	.....	5.2	445	13.0	1,070	3	106	.....	.....
7. Ballast	.....	.....	42.9	.....	43.5	.....	30	.....	48.5	.....
8. Total (1–7), approximate	180.5	6,500	180.5	6,500	180.5	6,500	180.5	6,500	180.5	6,500
9. Specific weight (2–6) without ballast for submerged propulsive installation	1.313		0.8		0.8		1.0		0.745	
10. Submerged installation based only on power developed and weight	43	1,151	8.6	116.5	32	1,430	22	458	.....	.....
11. Submerged installation dependent upon power per kg. per S.H.P. hour "b"	*35.8	960	7.47	100	18.35	820	18	376	.....	.....
12. Submerged installation dependent upon capacity only and weight	83.5	2,250	59.5	2,870	35.5	1,551	59	2,520	82	3,880
13. Submerged installation based on capacity per H.P. per hour "a"	†70 §40	1,870 1,060	27.75	1,435	29.9	1,135	3.69	157	.....	.....
14. Maximum capacity in E.H.P. surface propulsion	1,700		1,700		1,700		1,700		2,880 I.H.P.	
15. Maximum surface speed, in knots	16.5		16.5		16.5		16.5		17.7	
16. Submerged capacity in H.P. hours	*1,200 §2,100		2,000		1,370		16,000		1,700 to 2,160	
17. Max. sub. capacity considering loss of effective H.P. due to running submerged	1,200		1,150		1,740		1,220		1,440 I.H.P.	
18. Max. sub. speed, in knots	10.6		10.5		11.9		10.6		11.1	
19. Radius of action at max. speed in sea miles	10.6		18		9.4		138		13.1 to 16.7	
20. Submerged radius of action at 8 knots	28		26.8		18.2		214		22 to 29	
21. Time of sub. run at 8 knots, in hours	3.5		3.35		2.28		26.8		2.84 to 3.6	
22. Charging time, in hours	6 to 10		5		1.2		(‖)		2 to 4	

* One hour rate. † One hour discharge. ‡ Three and one-half hours discharge.
§ Three and one-half hours rate. ‖ From shore station.

N. B.—TRANSLATOR'S NOTE.—The original table gives space occupied in cubic meters, and these have been converted roughly into cubic feet in the above table.

Lines 1–6, inclusive, show the weight and space occupied by that installation when designed to come within the allowed space of 6,500 cubic feet.

Line 10 shows weight and space occupied if installation is restricted by weight alone (disregarding space occupied).

Line 17 takes into account the loss in effective H.P. running submerged, due to propeller inefficiency when propeller is designed for maximum surface speed.

THE SALVAGE OF THE U. S. S. *PRINCETON* AT PAGO PAGO, SAMOA.

By Ensign H. H. Little, Member.

SINKING.

At dusk on the evening of July 11, 1914, while returning to the Naval Station at Pago Pago from Massafau and Ava on the Island of Tutuila, American Samoa, the U. S. S. *Princeton* ran onto an uncharted pinnacle rock off Round Bluff (Fig. 1).

Round Bluff is about three miles East of Pago Pago harbor entrance, and about four miles from the steel dock at the Naval Station (Fig. 2).

The *Princeton* struck the rock, bow on; the heavy bronze stem casting was dented in about nine inches and cracked in several places in wake of the dent.

In scraping along over the rock, a section of the false keel, or shoe, was ripped off for a distance of forty feet back from the dent; the keel itself was gaged in several places.

From frame No. 26 to Frame 46 (Figs. 3 and 4), the jagged edge of the rock cut through the wooden hull, tearing off the copper sheathing, splintering and mashing the planking, and bending the steel frames on a line parallel to the keel, and at a distance from it of about five feet. Between frames 55 and 60, under fire rooms (Figs. 5 and 6) the hull was punctured slightly, showing that the sea probably lifted the ship off the rock at frame No. 46 and set her down again at frame No. 55 for an instant contact.

The official version of damage done, as found by the board of investigation, was as follows:

"The keel, directly abaft the forefoot, is shattered for a distance of about 12 feet, deepest indentation being about nine inches.

"On the port side there is a hole about 5 feet long by 4 feet wide between frames No. 36 and No. 39, upper edge of hole 8 feet below bilge keel.

"Frames Nos. 36 and 38 are buckled in at this hole about 18 inches. Forward of this hole, extending aft, the planking is cut and shattered from a section about 2 inches deep and 6 inches wide at frame No. 29, to about 4½ inches deep by 18 inches wide, at frame No. 36. Frames No. 30 to No. 36 are probably slightly buckled.

"Approximately the same condition exists running from frame No. 39 to frame No. 45 with frames No. 39, 40, 41 and 42 probably slightly buckled."

At the time of the accident the *Princeton* was making full speed—about 8 knots—for the Naval Station at Pago Pago.

Suddenly a terrific shock was felt, which jarred the ship from stem to stern. This was followed immediately by a grating, grinding feeling which ended in a second shock and jar; then all was still.

On the first impact the engines were backed. After a lapse of two and one-half minutes and after a hurried examination had been made, the engines were sent full speed ahead.

The first reports showed that water was spouting up through the hull into the hold (compartment A-6) and into magazine A-8. No other injuries were found at this time, so it was decided to make a run for the harbor.

The beach from Round Bluff to, and inside, the harbor, is fringed with coral reefs over which the seas break continually. There were, therefore, few places to beach the *Princeton* without endangering the lives of the crew in the breakers.

For this reason, and from assurances of the chief engineer that fires could be kept going for some time, it was decided to make a run for the inner harbor and the naval station dock.

The water was entering A-6 at a very rapid rate and, as this compartment was not provided with a watertight hatch, in less than six minutes from the time of the first impact the compartment was full and overflowing to the forward berth deck.

The dynamo room is located forward on this deck, so that it was soon flooded and the ship was then in total darkness.

On account of the high hills surrounding Pago Pago night comes on very rapidly, so that it was now quite dark.

All watertight doors and hatches had been closed, boats lowered to the rail and preparations made to abandon the ship.

In entering the harbor the greatest difficulty was experienced in passing Breaker Point, where an almost right-angled turn had to be made. The bow was down to the hawse pipes by this time and the stern was relatively very high; this gave the rudder but slight effect and the turn was made with great difficulty (Fig. A).

All available pumps were put on the main drain, but this was not sufficient to free the ship as fast as the water entered.

As the dynamos were submerged early, the work in the engine room was being done by candle-light only.

Soundings and observations on the way into the harbor showed that the water was gaining rapidly and that there was little hope of keeping the ship afloat. However, as long as the forward bulkhead, between the fireroom and the forward compartments held, there was reasonable hope of reaching the naval dock.

By this time a leak under the fireroom was discovered and the compartment was making water rapidly through the top of double bottom B-96. As the ship was now listing to port badly, the fires under boiler No. 2 were threatened.

To add to the excitement below, water was pouring through the tiller-rope holes to the top of both boilers, filling the fireroom with steam.

Fifty minutes after the *Princeton* hit the "Green Turtle," as the rock is called by the natives, she was alongside the dock at the naval station.

Attempts were made to get a collision mat over the hole, but without success. Then work was commenced on hauling the ship ahead to beach her in the shoal water ahead of dock (Fig. 7).

All the available natives, men and women, and the native U. S. guard of Fita Fitas, had turned to and were giving a hand with the lines and hand pumps, and some had even started a bucket line. All this availed nothing, and water by now had reached the forward gun deck and had worked its way aft through the non-watertight doors on that deck, until it poured down the ladder to the engine room.

The engines were used to assist in hauling ahead, but the cranks were turning over in water and fires had been hauled from under No. 2 boiler. With steam going down so pumps could not run, and with water gaining rapidly, the *Princeton* soon grounded in about twenty feet of water.

As the ship continued to settle she listed to port, or towards the shore, thus resting on and covering the gash cut by the rock. The ship finally came to rest at an angle of about 20 degrees to port.

The waterline forward was about half way up between the gun and berth deck, and aft water had not quite reached the deck.

As the *Princeton* was resting on the edge of a sloping reef all lines and cables available were run ashore to trees and deadmen to keep the ship from sliding off into deep water. On account of the few trees and their distance from the beach, holes had to be dug, and anchors, set in concrete, planted, to give additional holding power.

A diver was sent around the ship the first night, but he was unable to locate any injuries, as the ship was resting on the damaged spot.

During the night the ship's anchors were taken ashore and planted and additional lines run, so that the *Princeton* was secured with lines to all her bitts and masts.

While this was being done a four-inch pipe line with suction to wardroom was run from the power house, five hundred feet away, to the ship. This was completed at 8:00 A. M. the following morning.

Shortly after 8:00 A. M. a very peculiar and lucky accident occurred. Just as the power-house pump was started the

Princeton started to roll slowly to starboard; she broke lines and tore out anchors, but they so retarded her motion that the action was easy and the *Princeton* slid but a few feet, coming to rest with her starboard turn of bilges on the edge of the reef, and inclining away from the shore at an angle of about 40 degrees (Fig. 8).

The ship now rested on her starboard side with the port hammock nettings out of water at low tide. There was fourteen feet of water over the starboard netting.

The port side, where injury occurred, was thus left exposed, greatly facilitating the work of patching.

Work was again started to secure ship and keep her from sliding off the reef. Five-foot holes were dug and seven large concrete anchors were planted. The anchors were made up of large cast-iron conical bases, intended for pedestals for the steel dock, imbedded in concrete. The anchors weighed approximately two tons each. A six-inch hole through the center was provided for sections of anchor chain secured with a couple of half hitches. Shackles were bent to loose ends and lines run to 1¾-inch steel wire pendants secured to the mast heads. Three sheave blocks were placed in each line to take up the slack.

For this work the masts were strengthened by two 6-inch by 8-inch pieces placed as braces on each side of the masts. Also additional tackles and braces were run from mastheads down to the starboard nettings.

Work was then started to clear and lighten the vessel. Rifles, pistols and ordnance material was removed by divers. Guns were removed and sent by freight to Mare Island. Topmasts and topmast rigging and gaffs were removed. All deck hamper, including compass binnacles, wheels and pedestals, ventilator cowls, and bridge fittings were removed.

SALVAGE.

As soon as it was assured that the ship was secured and could not roll nor slide any more, divers were sent over to estimate the damage.

From their inspections it was believed that with certain gear, not then on station, the ship could be sealed up and raised.

A radiogram was therefore sent to the Navy Department for the following material:

Three 12-inch centrifugal pumps and necessary suction piping for same.

Ten No. 15 steel plates to plate inside for shoring.

Five hundred feet 6-inch × 6-inch lumber to be sized.

Two thousand feet 2-inch × 12-inch lumber to be sized.

Rivets, lag screws, oakum, copper sheeting, felt and hammock canvas.

While awaiting the arrival of this material work was commenced to systematically close all natural openings to the ship, such as hatches, skylights, ventilators, drains, gun ports, air ports, bunker vents, scuppers, voice tubes, et cetera. The main deck hatches were the only hatches closed.

The ships plans were taken and a pencil sketch made of all openings in the hull. The ship was marked off into sections and a diving party made responsible for the closing of every known opening in their assigned part of the ship. This work started the day following the sinking and continued for practically the entire period of 68 days that the ship was submerged.

The work of placing the main patch started July 30th, and was completed on September 1st, 32 days later.

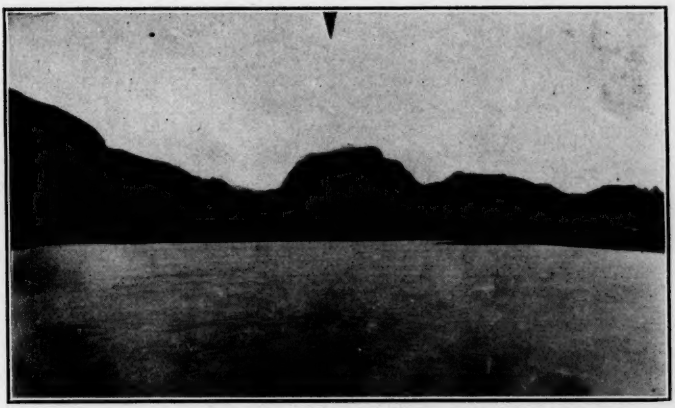
The work was done by divers working in pairs.

All copper sheathing surrounding the tear was ripped off first. Then the ship's side was smoothed down as much as possible.

A large single piece of sewed hammock canvas was first placed over the hole. This was tacked in place with sheathing nails. Over this a layer of 2-inch × 12-inch planks were spiked.

The job of placing the planks presented the greatest difficulty, as they were 24 feet long, had considerable buoyancy, and had to be sprung into place to follow the lines of the ship.

The matter of buoyancy was easily overcome. When a plank was needed below a large square steel bar, heavy enough



HARBOR OF PAGO PAGO AS SEEN FROM NAVAL DOCK, LOOKING TOWARD NARROW ENTRANCE ON RIGHT.

FIG. 1.—ISLAND OF TUTUILA, AMERICAN SAMOA. CROSS SHOWS ROCK STRUCK BY U. S. S. "PRINCETON."

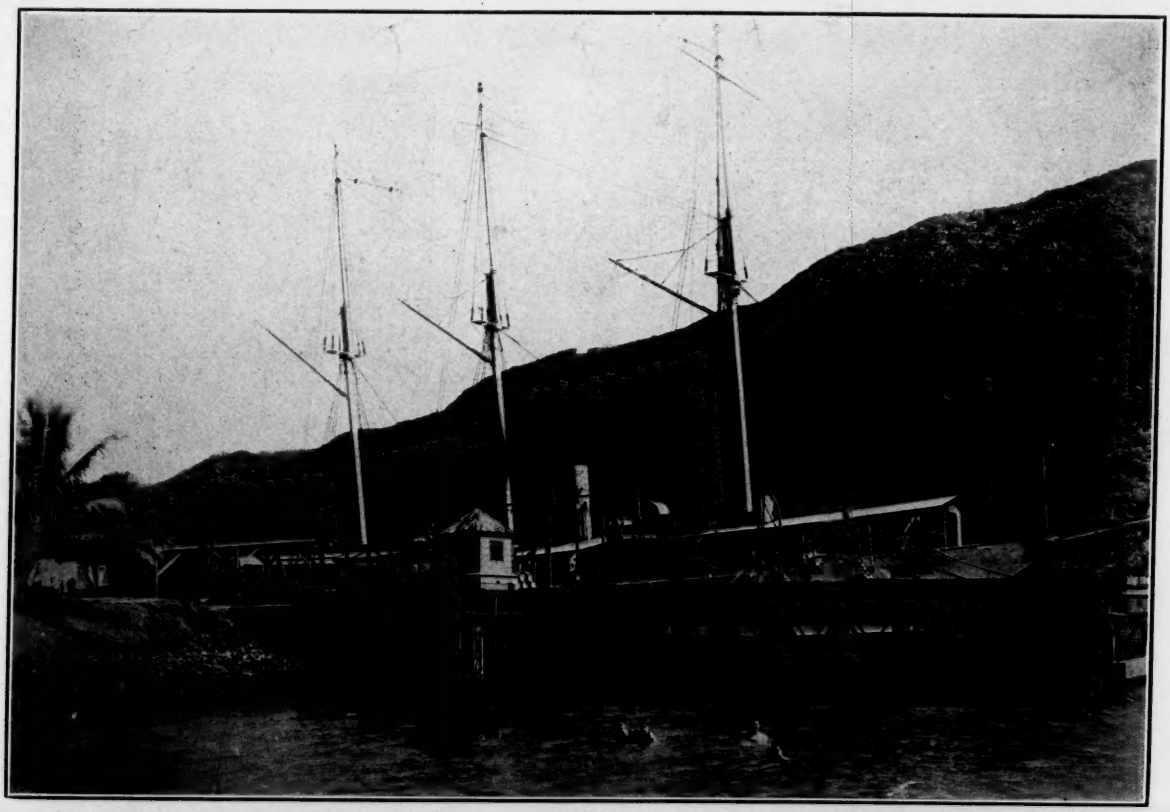


FIG. 2.—SHOWING U. S. S. "PRINCETON" AT STEEL DOCK, NAVAL STATION, TUTUILA, SAMOA.

FIG. 2.—SHOWING U. S. S. "PRINCETON" AT STEEL DOCK, NAVAL STATION, TUTUILA, SAMOA.

FIG 3.—SHOWING FORWARD SECTION OF PATCH.

FIG. 4.—SHOWING COMPLETE PATCH OF TWO LAYERS OF 2″ × 12″ PLANKS COPPERED.

FIG. 5.—SHOWING PORT SIDE UNDER FIREROOM.

Negative has been reversed, making this appear on starboard side.

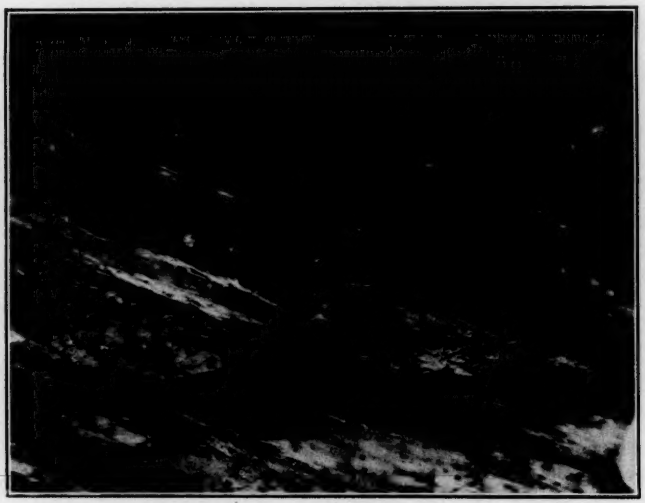


FIG. 6.—SHOWING GASH IN PLANKING UNDER FIREROOM.

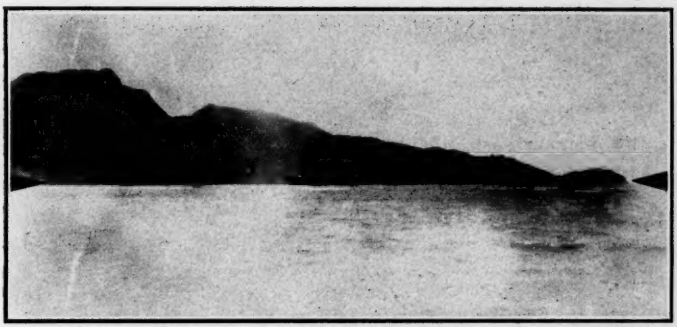


FIG. A.—SHOWING "OCEANIC" STEAMSHIP LEAVING HARBOR OF PAGO PAGO. BREAKER POINT ON EXTREME RIGHT.

FIG 7.—DOT AT ARROW SHOWS WHERE U. S. S. "PRINCETON" SANK.

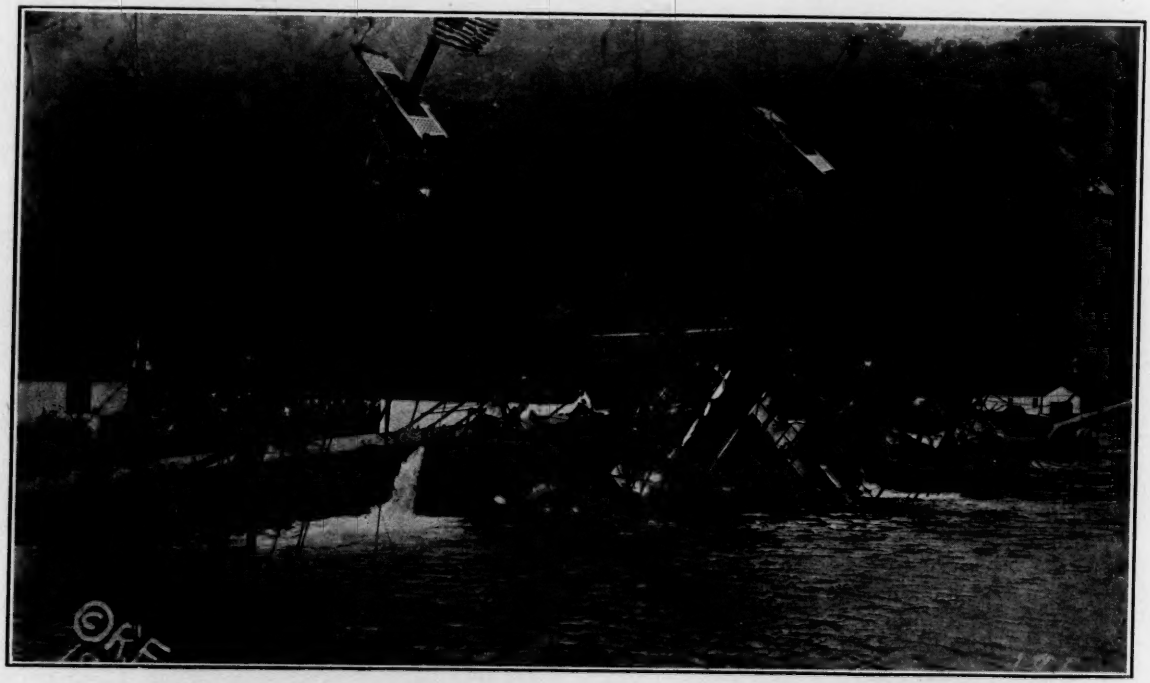
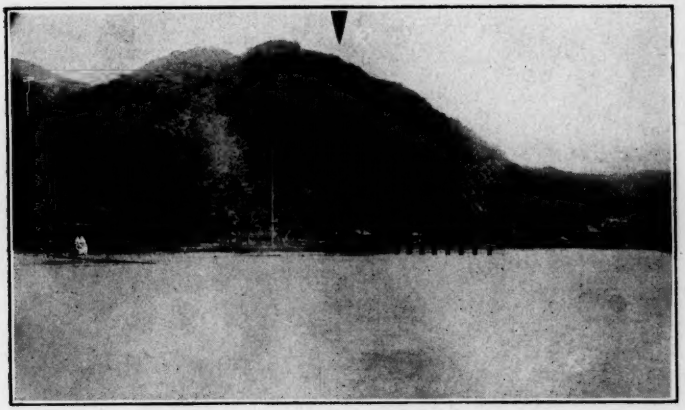
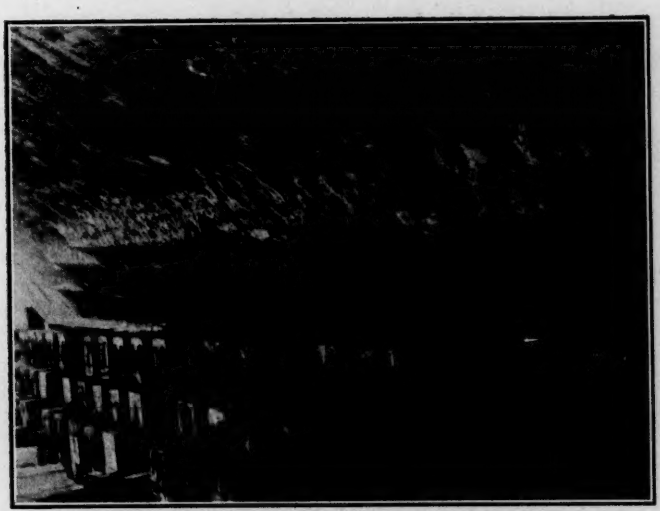


FIG. 8.—SHOWING U. S. S. "PRINCETON" ON BEACH AT NAVAL STATION. SHORTLY AFTER THIS PICTURE WAS TAKEN PRINCETON MOVED, AND ROSE SLOWLY TO AN EVEN KEEL AS THE WATER WENT DOWN.



ANOTHER VIEW OF HARBOR, ABOVE DOCK, SHOWING NAVAL ADMINISTRATION BUILDING ON LEFT.



ANOTHER VIEW OF FORWARD SECTION OF PATCH.

Actual condition at end of 6,000-mile cruise from Samoa to Fanning Island, to Honolulu, to San Francisco, to Puget Sound, plus 1,700 miles around Island.

to just sink the plank, was lashed to it. The divers would then straddle the plank and descend with it. When the plank was in place, with sufficient spikes to hold it, the iron bar would be cut loose and hauled to the surface for lashing to another plank.

A veritable workshop was maintained below the water at the ship's side. This contained spikes, sledges, caulking tools, augurs, hammers, etc., for the day's use.

As soon as a plank was secured in place divers would come to the surface for a short rest and would then descend with another plank. Divers used helmets only.

The time taken to lower a plank into place and secure it varied greatly, depending on the shape of the part of the ship to which the plank was spiked. Some single planks took up a whole day in placing; others only took two or three hours.

DESCRIPTION OF PATCH.

Two layers of planking were placed in all, one over the other, as shown in Figure 4. The first layer consisted of six planks butted, the seams between these planks and around the edges were caulked with oakum. The caulking of one layer of planking took one day.

Over the first layer of planking two layers of felt were tacked.

The second layer of six 2-inch × 12-inch planks were spiked over this as before, the second layer being placed to break joints with the first. Two more layers of felt were placed and the whole patch then covered with copper sheathing. It took just one week to cover the patch with copper.

Thirty-two days in all were spent on the main patch.

There was little delay on account of the weather, as the harbor of Pago Pago is well protected and the tide variation is slight.

The work of closing the interior openings of the ship went on simultaneously with the work on the outer hull. After

all small openings on the interior of the ship had been closed work was started on the main-deck hatches.

The hardest problem here was to make the engine and fireroom hatches and ventilators watertight. Stringers of 2-inch × 12-inch pieces were bolted to the sides of each hatch and the open space closed by 2-inch × 12-inch pieces, carefully caulked.

Ventilator cowls were removed and wooden covers of 2-inch × 12-inch pieces were wedged into these openings.

The umbrella opening around the stack was closed with canvas, but this carried away on first trial with the pumps, necessitating a wooden collar being wedged into the opening.

By the time this work had been completed the three pumps had arrived from the United States and the installation of these was started at once. One pump, in the condition received, was entirely unfitted for the work intended. It had a drum for electrical drive and there was no motor available on the island. The shaft was therefore cut off and a clutch fitted to gear with a steam-launch engine which was used. This worked satisfactorily.

Pumps were installed as follows: One was placed on shore, only to be moved aboard at a later period; this pump had a suction line leading to the engine room through the engine-room hatch. Steam for this pump was furnished by a donkey boiler installed on the deck near the original position of the pump.

The other two pumps were installed on platforms above the deck at the water line. One was placed at No. 2 hatch, with a suction leading down to A-6 (the hold). The other pump was placed at the port fireroom ventilator, with suction leading to fireroom floor plates. Two donkey boilers for these pumps were installed on platforms with pumps. The platforms were built at an angle to the deck, equal to about half the angle of the ship's list, so that when the ship was righted the boilers would be at approximately the same angle as when installed, but in the opposite direction.

Steam for the middle pump was supplied by the donkey boiler ashore, the pipe from boiler to pump being fitted with a grasshopper joint to allow for any movement when the ship came up.

As all the pumps were installed on platforms at an angle of about 20 degrees to 25 degrees to the deck, and as all suction piping entered hatches at right angles to the deck, it was a problem to make a joint between the suction pipes and the pump. These two connections came together at an angle. The problem was solved by collecting all the sheet lead on the station, soldering it together and making suitable elbows. The elbows were strengthened for the suction stress by spring brass-wire coils on the inside. The lead elbows were secured with ring flanges.

After all connections from suction lines to pumps had been made pumps were started in an attempt made to free the ship of water (Fig. 8). The water went down slowly for a distance of about six feet, when it stopped. Pumping was stopped and a search made for leaks. Several small ones, such as voice tubes, bunker vents and drains were found.

A few days later another attempt was made. Everything went smoothly, and the water went down rapidly until a seven-foot level was reached; then the water in the hull began to gain rapidly. Another search for leaks revealed the fact that the canvas covering around umbrella had been carried away. This was replaced with a wooden collar carefully caulked, and a third attempt made.

This final attempt was made on September 16. After pumping steadily for over five hours, with water going down gradually, the ship was seen to move slightly and roll more to starboard. A strain was taken on masthead pendants and all pumps slowed for an investigation. As everything seemed all right, the pumps were speeded up to full power again. When the water had lowered about eleven feet the ship started to come up. She rose steadily, and at the end of another hour, by continuous pumping and by hauling on masthead pendants, the ship was brought to an even keel.

A peculiar fact was noted as the ship came up—she rose in a vertical line, with no perceptible movements to right herself for a height of about fourteen feet, even though the masthead lines were being hauled upon. She then worked over gradually to an even keel.

While the ship was coming up divers were at work below opening up doors to give a free run for the water.

It took from noon of the 16th of September till about dusk to get the *Princeton* up and on an even keel. Pumps were kept running till eleven P. M., the shore pump having been moved aboard during the evening. Handy billies were then started and kept going all that night and all the next day, to drain compartments having no openings or drains to those containing suction lines.

The next day the midship pump was moved to the gun deck and a suction put into A-10, the forward coal bunker. The fireroom and engine rooms were pumped until there was only 10 inches of water left in them. By the second night the *Princeton* was up to very near her normal waterline.

Work was at once started on the fire and bilge pump. A steam line from boiler on dock was run to it, and 36 hours after the ship came up her own pump was running. All compartments were then drained.

A pump was also installed in the forward hold to handle the leakage there. It amounted to about one gallon per minute.

As soon as the space around the patch in compartments A-6 and A-8 was freed from water work was commenced on cementing over it. In order to fill hole with cement it was necessary to fill starboard side of bottom of compartment A-6, double bottom A-99, under magazine A-8, and a small amount in the double bottom under coal bunker A-10.

The sheet-metal top of double bottom A-99 had to be cut out in order to properly fill the space over the hold with cement. The cement was mixed on the berth deck and placed by bucketfuls, being worked into place around the jagged edges of the torn planking and the frames by hand. After cement had been

placed the tops were covered with 2-inch × 12-inch planks and the whole shored down with 6-inch × 6-inch beams. These were later tie-braced.

Natives, employed by the day, and the ship's force were put to work washing and scraping the paint work. All metal surfaces were scraped and painted. All wood furniture and fittings were torn out and renewed by ship's force. This included all stateroom furnishings, all office fixtures and portable wooden fixtures of all descriptions. Metal sheathing and insultation filling was all removed.

An old 220-V generator was sent from Mare Island and installed in place of the old machine, which was completely ruined. The ship was re-wired by the ship's force, the wires being run alongside of the old conduits and tapped to them.

In beaching, the *Princeton's* rudder had been left 25 degrees to starboard, so that when the ship rolled over to that side the weight of the ship came on the rudder, and the stock was twisted 15 degrees. As the stock was but slightly bent, the tiller simply had to be shifted.

After the ship came up the fifty feet of false keel which had been ripped off was replaced by divers, for docking purposes.

The engine plant was completely overhauled by the ship's force. All steel work on the pumps and engines was badly pitted. Engine piston rods and pump rods, valve stems and other small working parts of main engines and auxiliaries were taken up to the yard shop and a cut taken.

Nearly all pump cylinders were re-bored. The shaft was not lifted, but was jacked over and the bearings stoned. Impurities in the babbitt metal—probably zinc—were eaten out by galvanic action, giving it the appearance of worm-eaten wood. The bearings were not re-lined, but did not run hot.

The interior of the boilers, which had been full of salt water during the ship's submergence, showed little pitting or other corrosive effects. The combustion chamber of No. 2 boiler was slightly bulged, due, probably, to low water.

After the engines had been re-assembled the *Princeton* made

frequent trips around the island. In all she probably steamed 1,700 miles before her start back to the United States. During this time she had several breakdowns; in fact, she never made a trip that some minor breakdown didn't occur.

It might be mentioned here that, had the forward compartment, A-6 (the hold), had a watertight hatch, the *Princeton* could not have sunk.

By orders of the Navy Department the *Princeton* left Samoa September 25th, 1915, for the United States.

On arrival at the Puget Sound Navy Yard the ship was inspected by a Board of Inspection and Survey. This Board made recommendations that to put the vessel in first-class military condition would require an expenditure of $96,000, and this aside from the cost of a new battery.

For this reason the Department has under consideration a proposition to spend a small percentage of this sum, repair the hole in the hull, and turn the vessel over to the Naval Militia for inshore cruising.

POST GRADUATE EDUCATION IN NAVAL ENGINEERING.

By Lieutenant Commander John Halligan, Jr., U. S. N., Member.

The midshipman enters the U. S. Naval Academy at an average age of about 18 years. He has had as a rule the equivalent of a high-school education. During his four years at Annapolis he is prepared for the manifold duties of a naval officer, there being no attempt at specialization. The ground covered in the undergraduate course is necessarily extensive. In addition to this engineering training the graduate midshipman must be proficient in Navigation and well grounded in Gunnery, Seamanship, Naval Tactics, Military Tactics, International and Military Law, and in French and Spanish.

Engineering training is emphasized; by means of text books, revised from year to year, and well-equipped shops and laboratories, supplemented by practical work on board ship during the summer cruises, the midshipman is acquainted with the machines and appliances to be found on shipboard and with their method of operation. The hours assigned in the undergraduate course to subjects which pertain to naval engineering are as follows:

	Rec.	Prep.	Shop and Laboratory.
Marine engineering and naval construction....	442	589	138
Mathematics and mechanics	459	612	...
Electrical engineering, physics and chemistry..	340	453	16
Ordnance	153	204	81
Total for four years.....	1,394	1,858	235
Average per week	10.25	17.19	1.75

The scope of the engineering field with which the graduate midshipman must have some familiarity is so extensive that the undergraduate course is necessarily descriptive in its nature,

and time is not available for much fundamental theory. Even if time were available, it would probably be found unprofitable to give much instruction in theory at this early stage of the naval engineer's education. A better appreciation of theory comes after some experience has been had with its application.

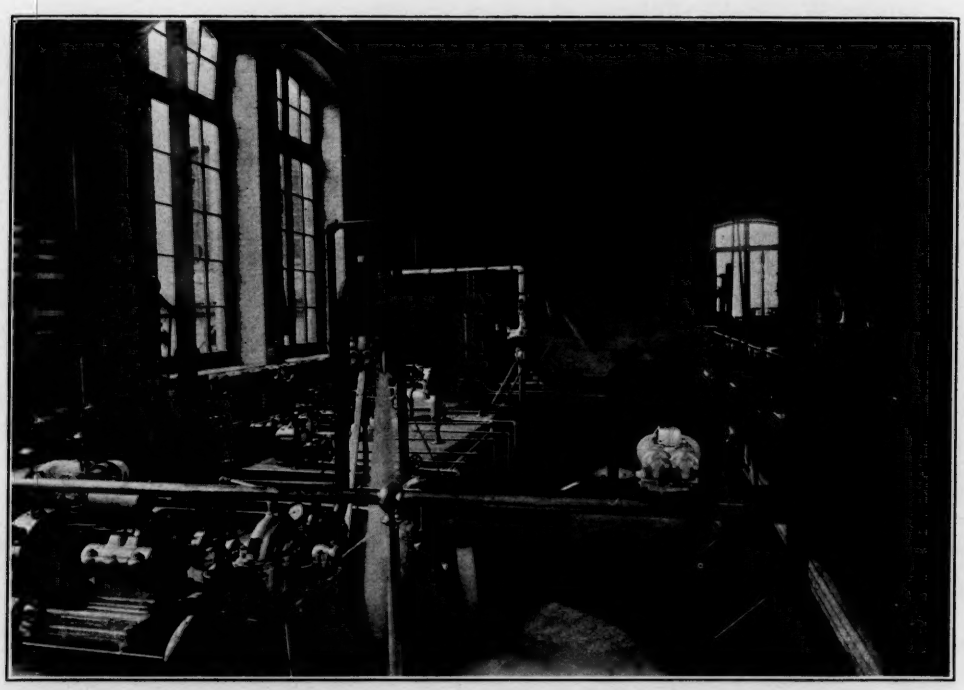
The young officer leaving Annapolis finds that his work lies largely with one or more of the various phases of naval engineering for 20 to 25 years until he obtains command rank. When as head of a department on board ship, as an inspector at a ship yard, as repair officer at a navy yard, or as a member of a technical bureau, he reaches a position of responsibility in connection with the operation, construction, repair or design of engineering material, the inadequacy of his undergraduate training becomes apparent. The function of the Post Graduate Department is to supply this deficiency in engineering education and training.

SCOPE OF THE FIELD OF NAVAL ENGINEERING.

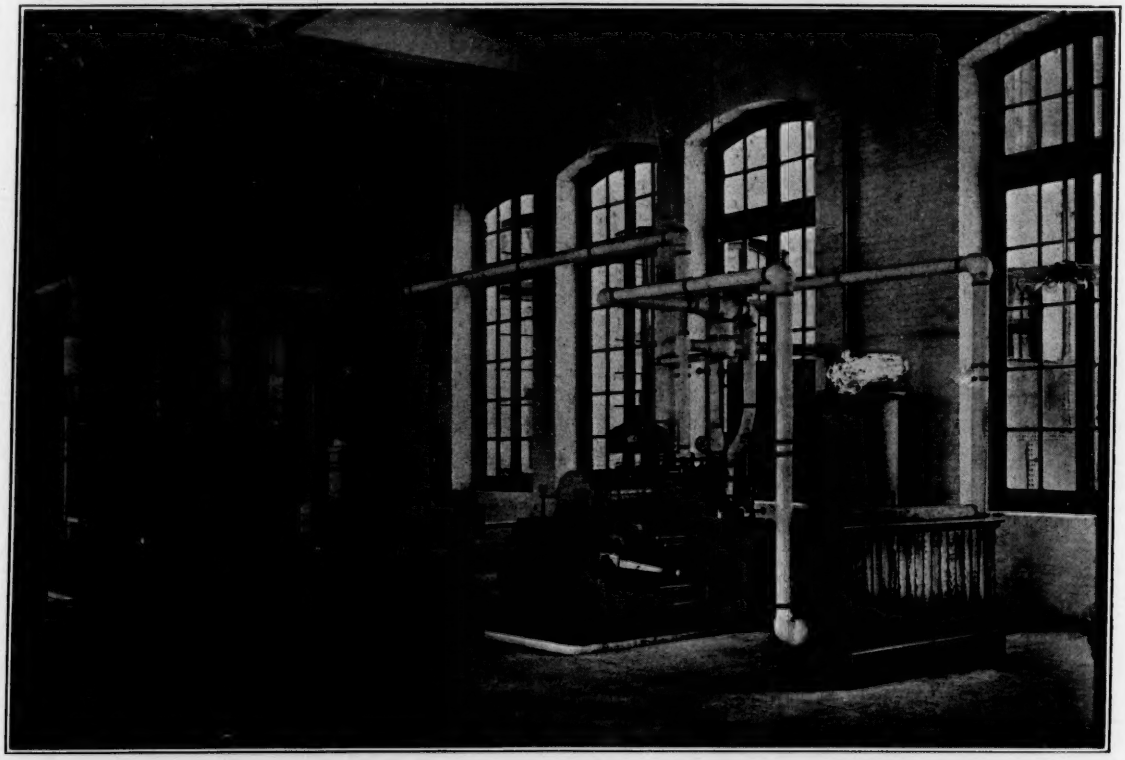
The technical duties in the various phases of naval engineering for which post graduate education is required are:

Hull Engineering (naval construction).—The designing, building, fitting and repairing of hulls of ships, capstans, windlasses, steering gear and ventilating apparatus; the designing, construction and installation of certain ammunition hoists; the placing and securing of armor; the designing and installation of supports for everything attached to the hull of the ship, including armament and propelling machinery, electric turret-turning machinery, boat cranes, deck winches; the docking of ships.

Ordnance Engineering.—The designing, manufacture, purchasing and inspecting of guns and gun mounts, ammunition hoists, rammers, gun sights and telescopes, periscopes and other optical instruments, smokless powder, gun cotton and other high explosives, torpedoes and mines, torpedo tubes, air compressors, armor, projectiles, small arms and infantry equipments, fire-control apparatus.



HYDRAULIC TANK.—ENGINEERING LABORATORY.



VIEW IN ENGINEERING LABORATORY.

View in Engineering Laboratory.

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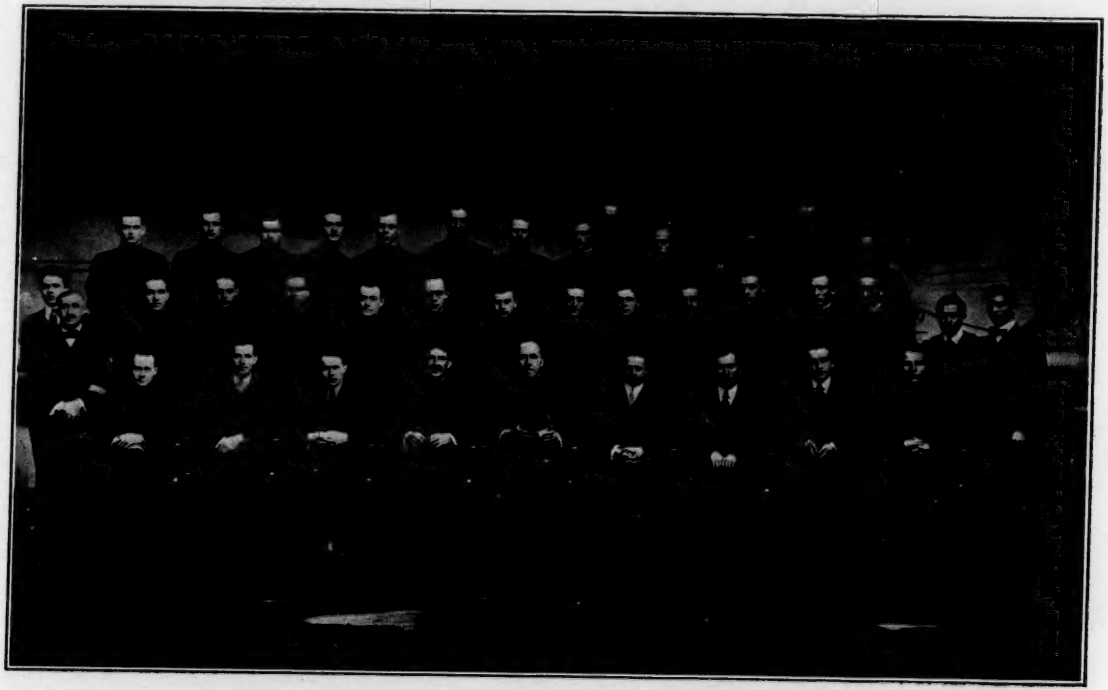


CLASS OF 1916, POST GRADUATE DEPARTMENT.

The management of the Naval Gun Factory, the Naval Proving Ground, the Naval Smokeless-Powder Factory, and various naval magazines.

Electrical Engineering.—At sea.—As electrical officers on shipboard, the operation and maintenance of all electric machinery, from the switchboard.

On shore.—The designing, purchasing and inspecting of all electric machinery and appliances except turret-turning motors, ammunition-hoist motors, boat cranes and deck winches.

Radio Engineering.—At sea.—As radio officers on staffs of commanders-in-chief, squadron and flotilla commanders; development and control of radio communication and maintenance of apparatus.

On shore.—Designing, inspecting, fitting and repairing of radio installations. The administration of the naval radio service.

Marine Engineering.—At sea.—As chief engineers or assistant engineers, the operating, maintenance and repair of motive and auxiliary machinery.

On shore.—The designing, building, purchase, inspecting and repairing of propelling machinery, boilers, engines, pumps, distilling apparatus, refrigerating apparatus, dynamo engines. The management of the Engineering Experiment Station and of the Fuel-Oil Testing Plant.

Civil Engineering.—The designing and constructing of buildings and their permanent fixtures at navy yards and naval stations; harbor works; waterfront improvements; graving docks and floating docks; power plants; roads, streets, tracks; hoisting appliances, cranes, derricks; transporation equipment; steam shovels, dredges and pile drivers; fire apparatus.

HISTORY OF NAVAL POST GRADUATE EDUCATION.

For nearly thirty years naval constructors have been given post graduate courses, generally of three years, at technical schools in England, France and Scotland, and more recently

at the Massachusetts Institute of Technology. The Bureau of Ordnance has, since about 1905, instructed from three to five men annually for periods of one or two years at the steel plants, the Naval Gun Factory, and the Naval Proving Ground.

Until within the past few years the naval civil engineers were appointed from civil life. They were, of course, graduates in civil engineering.

The necessity for post graduate instruction in electrical and in marine engineering has been recognized sporadically during the past three decades, but the demand for it has not been insistent except during the past fifteen years.

Prior to 1900 naval propelling machinery was well standardized. The reciprocating steam engine was the only prime mover above the horizon. The undergraduate education at Annapolis, supplemented by training at sea, sufficed for the comparatively simple demands on engineer officers.

At present we are employing seven distinctly different types of propelling machinery; these are: Reciprocating steam engine; Parsons turbine, direct drive; Curtis turbine, direct drive; electric propulsion; turbines with mechanical reduction gear; Diesel engines; gasoline engines.

Producer-gas plants, hydraulic reduction gear, and combinations of the foregoing types have been seriously considered for naval use.

During this same period the electrical installations on shipboard have developed from insignificant to important dimensions.

To prepare the engineer officer to cope with and to assist in the rapid development of naval machinery, there is now required a thorough training in the fundamental theory of thermodynamics, machine design, strength of materials, metallurgy, electricity, applied mechanics and engineering mathematics.

It is important that in the post graduate work fundamental theory be emphasized, because this is the naval officer's only opportunity to acquire theory. During the greater part of his active career he will be associated with engineering practice.

ORIGIN OF PRESENT POST GRADUATE DEPARTMENT.

About 12 years ago a class in marine engineering, consisting of about 6 students, was organized in the Bureau of Steam Engineering. These men remained in the Bureau pursuing a course of reading under the guidance of an officer attached to the Bureau. They accompanied the Trial Board on acceptance trials of new ships, taking performance data of the machinery. In 1909, a School of Marine Engineering was established at the Naval Academy with a class of 10 men with experience of from 4 to 11 years since graduation. The course then established was one of two years' duration, and consisted of discretional reading by the student under the guidance of the head of the department, a tour on the Bureau of Steam Engineering trips during the summer to manufacturing plants. Each student was allowed to choose a specialty under engineering, and a technical library was accumulated with the idea of affording him access to all that had been written on the subject of his specialty.

The students found that they were hampered in their reading by the inadequacy of their education in mathematics, mechanics and thermodynamics. Notwithstanding this handicap much good was accomplished in thus directing the attention of selected men to engineering. The subsequent engineering service of this and succeeding classes has been so conspicuously valuable as definitely to establish the importance of post graduate education.

To meet the demand for training in fundamental theory a system of lectures was introduced into the course. The services of distinguished engineers and educators were obtained, some of whom delivered series of lectures extending over periods as long as four months. These lectures were excellent in character, and certain of them have been published and afford valuable contributions to engineering literature. An effort was made to direct the reading of the students to the end that they would prepare themselves for the subjects covered by the lecturers. In many cases, however, it was

found that this preparation was futile, in that there was insufficient control of the treatment of subjects by the lecturer, and the students found that the subject of the lecture was not that for which they had prepared themselves. It was natural, therefore, in the evolution of the school, that there should be established resident professors with a definite schedule of lectures, laboratory work and preparation.

In 1912 the title of the school was changed from "School of Marine Engineering" to "Post Graduate Department, U. S. Naval Academy," and its field was enlarged to include the directing of the post graduate work of the naval constructors, ordnance, electrical, radio and civil engineers.

In the course of development of the Post Graduate Department there has been much discussion and diversity of opinion as to where it might best be located and as to the extent to which the facilities of colleges and universities should be used. The predominant service opinion at present is that the headquarters should be at Annapolis, and that after one year's work here the men should be sent to universities not under naval control, for the purpose of broadening their viewpoint by contact with civilians. It seems important to maintain the first part of the course under the direct control of naval officers, in order to permit close observation of the students with a view to determining their fitness for subsequent post graduate training, and because in the beginning it is important to indoctrinate them as to their future work. Other considerations are the availability of the well-equipped laboratories of the Naval Academy, and the proximity of the Naval Engineering Experiment Station.

The present direction of development of the Post Graduate Department is toward providing in the first year at Annapolis common courses of high efficiency for all groups in the fundamentals of engineering.

Courses which seem thus commonly to be required are: Mathematics, mechanics, and applied mechanics; electricity; heat engineering; machine design; strength of materials;

metallurgy, including the chemistry of steels and bronzes, and metallography; engineering laboratory; electrical laboratory.

With the exception of metallurgy, all the foregoing courses have been established.

THE PRESENT SCHOOL.

The Post Graduate Department, under Article 1,543, U. S. Naval Regulations, offers special training along the following lines:

Hull engineering (naval construction), general.

Ordnance engineering, general and

(a) Gun Design,
(b) Optics,
(c) Explosives,
(d) Metallurgy.

Electrical engineering, general.

Radio engineering.

Marine Engineering, general, and

(a) Design,
(b) Shop Practices,
(c) Metallurgy,
(d) High-speed internal-combustion motors for aeroplanes,
(e) Diesel engines, with their application to submarines.

Civil engineering.

The location of the training of the various groups is given in Table I.

TABLE I.

Courses.	At Annapolis.	Subsequent Training.
Naval Construction...	7 mos., Feb. to Sept...	2 years at Mass. Ins. of Tech. leading to degree of M. S.
Ordnance.....	4 mos., Oct. to Feb....	6 mos., Naval Proving Ground. 4 mos., Steel Plant. 4 mos., Naval Gun Factory. 1 mo., Bausch & Lomb, Rochester, N. Y. 1 mo., Sperry Gyroscope Co., Brooklyn, N. Y. 4 mos., additional specializing at one of the foregoing.

Courses.	At Annapolis.	Subsequent Training.
Electrical	1 year	1 year, Columbia University, leading to degree of M. S.
Radio		Summer of 2d year at Navy Yard, N. Y.
Marine Engineering		Schenectady, N. Y. Sperry Gyroscope Co., Brooklyn, N. Y.
Civil Engineering	1 year	2 years at Rensselaer Polytechnic Institute, leading to degree of C. E. Summer work under direction of Bureau of Y. & D.

The administration of these courses is governed by a council consisting of the Superintendent of the Naval Academy, the head of the Engineering Experiment Station, the heads of the academic departments of marine engineering and naval construction, ordnance and gunnery, electrical engineering and physics, mathematics and mechanics, a naval constructor, a civil engineer, and the head of the Post Graduate Department.

The head of the Post Graduate Department is an officer of engineering experience.

The staff of instructors at Annapolis is as follows:

		Per annum.
H. A. Everett, Professor of Marine Engineering.	S. B., Massachusetts Institute of Technology.	$3,500
R. E. Root, Professor of Engineering, Mathematics and Mechanics.	S. B., Morningside College M. S., State University of Iowa. Ph. D., University of Chicago.	3,000
L. A. Doggett, Professor of Electrical Engineering.	A. B., Harvard University M. E. E., Harvard University.	3,000
J. G. Russell, Instructor in Engineering Subjects.	S. B., Massachusetts Institute of Technology.	1,800
H. B. Lindsay, Instructor in Physics.	S. B., Worcester Polytechnic	1,800
H. E. Jenks, Instructor	S. B., Worcester Polytechnic	1,800

The courses in chemistry are conducted by Prof. of Mathematics Paul J. Dashiell, U. S. N., who has charge of the undergraduate work in chemistry.

With the exception of Prof. Dashiell, none of the foregoing

has any duty in connection with the undergraduate work at the Naval Academy.

Typical time schedules of the work at Annapolis are shown in Tables II, III and IV.

TABLE II.—ENGINEERING—FIRST TERM.

For Marine Electrical, Radio and Civil Engineers.

	9	10	11	12	1	2	5:30
Mon.	Thermodynam. lecture.	Study.	Mechanics lecture		Gym.		Marine engine design. Drafting room.
Tues.	Mathematics. Lecture A. Study B.	Lecture B. Study A.	Chemical laboratory.				Electrical laboratory.
Wed.	Electricity lecture.	Study.	Steam mach. lecture.		Gym.		Engineering laboratory.
Thurs	Mathematics. Lecture B. Study A.	Lecture A. Study B.	Chemical laboratory.				Engineering laboratory.
Fri.	Thermodynam. lecture.	Study.	Electricity lecture.		Gym.		Electrical laboratory.
Sat.	Study.	Applied mechanics	Laboratory reports.				

Assignment of Hours per Week to Subjects.

	Lecture or exercise.	Laboratory.	Preparation.
Mathematics	2	...	6
Mechanics	1	...	2
Applied mechanics	1	...	2
Thermodynamics	2	...	4
Steam machinery	1	...	2
Marine engine design	3½	...	1
Engineering laboratory	...	7	...
Electricity	2	7	4
Chemistry	...	4	...
Gymnasium	3	...	...
Totals	15½	18	21

NOTE.—For the 21 hours preparation per week, 6 hours are available in class room.

TABLE III.—POST-GRADUATE DEPARTMENT.

NAVAL CONSTRUCTION.

Spring Term.

	9	10	11	12	1	2	5:30
Mon.	Mechanical lecture.	Study.	Heat Eng'g lecture.	Gym.			Laboratory reports.
Tues.	Mathematical lecture.	Study.	Heat Eng'g lecture.	Warship design.			Warship design.
Wed.	Mechanical lecture.	Study.	Electricity lecture.	Gym.			Electrical laboratory.
Thurs	Mathematical lecture.	Engineering laboratory.					Chemical laboratory.
Fri.	Mechanical lecture.	Study.	Chemistry lecture.	Gym.			Chemical laboratory.
Sat.	Mathematical lecture.	Engineering laboratory.					

Hours per Week.

	Lecture or exercise.	Laboratory.	Preparation.
Mathematics	3	...	6
Mechanics	3	...	6
Heat engineering	2	...	4
Chemistry	1	7	2
Warship design	4.5	...	...
Electricity	1	...	2
Electric laboratory	...	3.5	1.5
Engineering laboratory	...	6	2
Gymnasium	3	...	...
	17.5	16.5	23.5
Preparation hours in class			7.5
Home work			16

TABLE IV.—POST-GRADUATE DEPARTMENT.

Schedule for Ordnance Group.

First Term.

	8	9	10	11	12	1	2	4:30
Mon.	Ballistics.			Mechanics. Lecture. / Study.	Gym.			Ballistics. Study.
Tues.	Mathematics. Study / Lecture.		Study	Metallurgy. Engineering laboratory.				Chemistry.
Wed.	Metallography.			Ballistics.	Gym.			Engineering laboratory.
Thur.	Mathematics. Study / Lecture.		Study	Chemistry.				Metallography.
Fri.	Mechanics. Study / Lecture.			Ordnance optics.	Gym.			Mechanics. Study.
Sat.	Mathematics. Study / Lecture.			Metallography.				

Assignment of Hours per Week to Subjects.

	Lecture or exercise.	Laboratory.	Preparation.
Mathematics	3	0	8
Mechanics	2	0	5
Mechanical engineering laboratory	0	2½	1
Metallurgy	0	2	1
Metallography	0	7½	3
Chemistry	2	2	2
Ballistics	4	0	4
Ordnance optics	1	1	2
Gymnasium	3	0	0
Totals	15	15	26

Detailed descriptions of the courses at Annapolis and elsewhere are contained in the Catalogue of Curricula and Courses of Study, of which a limited number of copies are available to officers who are interested in post graduate training.

In this paper space permits only a general discussion of certain features of the work.

NATURE OF THE WORK.

Someone has said that engineering is one part arithmetic and two parts common sense, and that of these the common sense is the more difficult to acquire. Naval post graduate education aims to develop this quaility of common sense in the student. He is taught to think logically and his judgment is trained.

Some difficulty is experienced in weaning the students from the habits of memory study for marks, which they have acquired as midshipmen. Marks are not posted; each student is permitted to see his own, but is not acquainted with those of his classmates. The men are encouraged to study for what they can get out of the subjects, rather than for marks.

It will be noted from the time schedules (Tables II, III and IV), that the forenoons are generally devoted to lectures, or more properly recitations, and the afternoons to drafting room or laboratory work, the division of time between theory and practical work being about even.

There is scheduled about 55 hours work per week (lecture, laboratory and preparation). The home work required is about 15 hours per week,—the men average 20 hours.

It has been found necessary to establish a control of the amount of preparation work assigned by the professors. This is secured by means of weekly reports from the students, detailing the time spent in the preparation of each subject. The reports are anonymous, and are therefore frank.

An important part of the course is the gymnasium work, to which there are assigned three periods a week, before luncheon. In each period there is 15 minutes of Swedish exercises, during which all muscles are flexed, followed by 45 minutes of hand ball. A hand-ball league has been organized with scheduled games for teams comprising all the students and instructors. Twenty-four players are accommodated at one time in the courts available. When the weather permits, tennis will replace hand

ball; for this there are 24 courts available. The purpose of this scheduled exercise is to maintain the physical health of the students and to teach them forms of exercise which they may take to sea.

CHARACTER OF STUDENTS.

Selection of officers for post graduate instruction is made by the Navy Department from applicants of best service record, on recommendation of the technical bureaus concerned. The present requirements of sea service since graduation from the Naval Academy are, for the naval constructors and civil engineers, two years; for all other groups, five years.

The Naval Post Graduate student is unusual in the following respects:

1st. He has, in many cases, been chosen for post graduate instruction because of aptitude in the practical application of the theory which he is subsequently to study.

2d. He is ambitious and zealous. With very few exceptions he is disposed to work too much rather than too little.

3d. His time as a student is more valuable than is that of his instructors. The shortage of officers in the Navy is such that when an officer is detached from a vessel for purposes of post graduate instruction he leaves a vacancy which cannot be filled. While a student he receives the pay of his rank, which is about $2,400 per annum.

Except in the cases of naval constructors and civil engineers, the services of the officers for post graduate instruction can be spared for a period of only two years. It has been necessary, therefore, in naval post graduate education, and particularly in that part of the course which is conducted here, to insist on high efficiency of instruction. The professors and instructors are selected with care and they are employed in sufficient number to permit individual instruction to the greatest possible extent.

NUMBER OF STUDENTS.

Officers under instruction in the various courses in January, 1916, are as follows:

At Annapolis:
16 Marine engineers;
4 Electrical engineers;
1 Radio engineer;
5 Ordnance men;
5 Naval constructors.

At Columbia University:
7 Marine engineers (design option);
2 Marine engineers (shop practices option);
10 Electrical engineers;
1 Metallographist.

At Massachusetts Institute of Technology:
13 Naval constructors.

At Rensselaer Polytechnic Institute:
3 Civil engineers.

At Steel Plants, Gun Factory, Naval Proving Ground, etc:
9 Ordnance engineers.

The number of naval construction and of civil engineering students is governed by the prospective vacancies in those corps.

Of the ordnance, marine engineering, electrical and radio groups, the numbers are determined by the requirements of the service, insofar as they can be met in view of the existing shortage of officers for duty on shipboard.

The Bureau of Steam Engineering's statement of existing conditions in this respect is as follows:

"The Navy, as constituted and organized on January 1, 1916, required for efficient operation the following number of Post Graduate Engineer Officers below the grade of Commander:

Fleet, Squadron and Flotilla Engineers	6
Battleships, First Line (7)—Engineer, Assistant, Electrical	21
Battleships, Second Line (18)—Engineer, Assistant	36
Armored cruisers (5)—Engineer, Assistant	10
Cruisers (all classes) (13)—Engineer Officer	13
Destroyers (38)—Engineer Officer	38
Submarines (37)	20
Inspectors and Assistant Inspectors of Machinery	12
Inspectors and Assistant Inspectors of Engineering Material	6
In connection with ships building	11
Navy Department	14
Navy Yards and Stations	36
Total	223

"There are fifty-six (56) post graduates in the service and two hundred and twenty-three (223) are needed excluding the requirements of the ships building."

NOTES.

MARINE PROPELLING MACHINERY.

TURBO-ELECTRIC PROPULSION.

Turbo-electric propulsion is being developed in Sweden by the Svenska Turbinfabriks Aktiebolaget Ljungstrom, of Finspong, who have installed 180,000 H.P. on land and have recently fitted a steamer, the *Mjolner*, with their turbo-alternators supplying current for two motors geared to the propeller shaft, steam being supplied from cylindrical boilers fitted with superheaters. The vessel is 225 feet by 36 feet by 15 feet 6 inches, with a displacement of 2,250 tons and a block coefficient of .66. The speed is 11.8 knots on 900 I.H.P. The normal output of the turbines is equal to 900 I.H.P., the motors developing 450 H.P. each. These motors drive on to a single propeller shaft through gearing, the reduction being from 800 r.p.m. at the motors to 85 r.p.m. on the propeller shaft. The boilers work under forced draft, and with 250 degrees F. superheat and coal having a calorific value of approximately 14,000 B.T.U.'s; the coal consumption for all purposes has been ascertained on service to be at the rate of .89 pounds per hour per I.H.P. Comparative trials carried on with a sister vessel, the *Mimer*, fitted with triple-expansion engines indicating 900 I.H.P., and the same quality of coal showed a consumption at the rate of 1.54 pounds per hour per I.H.P. for all purposes. In terms of coal consumption, therefore, the Ljungstrom turbo-electrical installation is the more economical by no less than 42 per cent., in addition to which the total weight of the Ljungstrom propelling machinery in the *Mjolner*, including boilers, is 74 tons less than the total weight of the machinery in the triple-expansion-engined *Mimer*.

The turbines in the *Mjolner* are of the Ljungstrom reaction type, running at 7,200 revolutions, arranged for radial flow of the steam through concentric blade rings fixed on two turbine discs. Each disc runs at the same number of revolutions, in opposite directions, so that without increasing the absolute circumferential speed the relative speed becomes 14,400 revolutions. In effect, the blades of the one turbine disc act as guide plates for directing the steam flow against the blades of the other turbine disc, there being no fixed blading as in the usual reaction turbine. The great capacity of the Ljungstrom machine for high extraction of energy with small weight and dimensions of parts is due to the relative velocity between blades and guide blades being twice as high as in the case where the guide blades remain stationary. There is also a marked reduction in leakage losses. According to this system of radial flow, the live steam enters the turbine at the center of the rotating-blade rings and expanding through the concentric revolving-blade wheels, passes finally to the condenser, which, in the *Mjolner*, is of extra high vacuum, the air pump being rotary design. It is claimed that, owing to the ingenious arrangement of the attachment of the rotating blades to the turbine discs rotating through expansion rings, the Ljungstrom turbine may be used with the highest steam temperatures and steam pressures attainable with the existing types of boilers and superheaters. Similarly, the exhaust temperature can, with great advantage, be carried down to the limit cor-

responding with the highest of the engine room. In the *Mjolner* the turbo-alternators are mounted directly on the condensers, and the appearance of the sets in the engine room is more suggestive of auxiliaries than of main propelling units, being located on the starboard and port sides of the engine room. This system of turbo-electric propulsion has already been taken up by British builders, who guarantee for an installation of 1,200 H.P. a coal consumption at the rate of .9 pound per I.H.P. hour (28 inches vacuum at 200 degrees F. superheat) inclusive of all auxiliaries. This latter installation embodies double reduction, being mechanically geared, with a reversing idler pinion to counteract the counter rotation of one element of the turbine rotor.—"Shipping Illustrated."

SOME COMPARISONS RELATING TO ELECTRIC PROPULSION OF A BATTLESHIP.*

The Navy Department of the United States has awarded to the General Electric Company of Schenectady, N. Y., a contract for electric propelling machinery for the new battleship *California*, which is being built at the New York Navy Yard. This ship is of the largest and most powerful class which has been adopted by the United States. Her displacement is 32,000 tons and her maximum speed is to be about 22 knots, requiring about 37,000 S.H.P. The contract with the General Electric Company covers two turbine-driven generating units, four propelling motors (one for each shaft), switching apparatus, cables, instruments, &c., two turbine-driven exciting units, and a complete equipment of condensing auxiliaries and ventilating blowers, all driven by motors from the exciting units. In fact, it covers practically the entire engine-room equipment except the main condensers.

Each of the auxiliary units is of 300 kw. capacity with a 240-volt direct-current generator geared to a high-speed, non-condensing turbine. These turbines will exhaust into the heaters, into the main turbines, or both.

The motors which drive the auxiliaries will be designed for a considerable range of speed variation so that the auxiliaries will be adaptable to economical conditions at different speeds of the ship.

The award of this contract was the result of a very long campaign of education which has been conducted by the writer, in which the equipment of the collier *Jupiter* has been an incident. The writer has from the first maintained that the proper field for electric propulsion was in ships of high power and in ships where economy at widely varying speeds was important.

The *Jupiter* has now been in commission for two years and has proved an unqualified success. The steam consumption of the main drive with 190 pounds pressure, dry steam, and 28.5 inches vacuum is 11 pounds per H.P. hour delivered to the propeller shafts, which is at least 30 per cent. better than is done by reciprocating engines in ships of the class. There is also evidence of further saving in the *Jupiter*, incident to the fact that her large low-speed propellers are turned with a perfectly uniform angular motion and with entire freedom from racing, conditions heretofore unknown with such propellers.

The generators for the *California* are bipolar alternators, and the motors are arranged to be connected either for 24 poles or 36 poles. For economic cruising at 15 knots or less, one generator will be used with motors on 36-pole connection. For higher speeds the 24-pole connection

*Abstract of a paper read by Mr. W. L. R. Emmet, Member of Council, at the 23d general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 18 and 19, 1915.

will be used. The ship will be capable of operating at a speed of about 18.5 knots with one generator.

Speed variation with either motor connection will be effected by change of turbine speeds through the agency of variable-speed governors designed to hold automatically any desired speed within usual ranges. This arrangement, which is also used in the *Jupiter*, entirely prevents racing and makes it convenient to hold a fixed speed irrespective of variations in sea, weather or steam conditions. It is thought that this feature will be valuable in fleet formations.

The steam consumption guaranteed on the *California* covers the total steam required for the main turbines and engine-room auxiliaries as described above. The conditions are 250 pounds gage pressure, dry steam, with such vacuum as can be produced under trial conditions. The guaranteed water rates per horsepower delivered to the propeller shafts are as follow:

	Pounds.
At 10 knots	14.6
At 15 knots	11.4
At 19 knots	11.1
Maximum speed	11.9

Very heavy penalties are imposed in case these guaranteed consumptions are exceeded in trials, $25,000 per pound for the two lower speeds and $20,000 per pound for the two upper speeds.

At full speed the *California* propellers will make 175 r.p.m., this being about the lowest speed of propellers which is practicable within the space. The propeller speeds proposed for the sister ships with Parsons turbine drive is 240 r.p.m., and comparisons by Dyson's method indicate that this speed difference will give the *California* an advantage of about 9 per cent. in propeller efficiency.

The following table gives a comparison of steam consumption per effective horsepower between the *California* as guaranteed, the *Florida* and *Utah* which are driven by Parsons turbines, and the *Delaware*, which is driven by reciprocating engines. These figures are taken from the published records of trials.

POUNDS OF STEAM TO MAIN ENGINES PER HOUR PER EFFECTIVE HORSEPOWER.

	12 kts.	15 kts.	19 kts.	21 kts.	Prop. speed. 21 kts.
Florida	31.8	—	24.0	23.0	328
Utah	28.7	—	20.3	21.0	323
Delaware	22.0	—	18.7	21.0	122
California	17.3	15.2	15.0	16.4	175

Such a table as this affords the truest basis of comparison of prime movers in ships.

The estimated weight of the propelling machinery without condensing auxiliaries is 530 tons, and that of the turbines originally proposed was 653 tons.

Comparing the arrangement of the machinery of the *California* with the *Utah*, and a geared turbine equipment for a large warship shown in a paper by Sir Charles Parsons read in March of 1913, Mr. Emmet drew attention to the simplicity of piping, steam and vacuum connections with the electric drive as compared with the other methods. The *California* has two turbines, the *Utah* has 10, and the proposed Parsons geared equipment has 32.

All of the steam passages between turbines in these arrangements in-

volve loss of useful pressure, and many of them involve risk of vacuum loss through air leakage, for it must be remembered that ships are often subject to vibration and that there may be appreciable relative motion of these different parts through expansion strains or working of the ship, or through their own inertia when the ship is vibrating or laboring in a seaway.—"Shipbuilding and Shipping Record."

DIESEL ENGINE PROPELLED SHIP *PACIFIC.*

By W. Kaemmerer.

The Burmeister and Wain Company, of Copenhagen, are the largest builders of Diesel engine ships. They now have under construction ten ships of 10,000 tons each, three of 9,000 tons, six of 8,000 tons, one of 7,000 tons, and three of 6,500 tons.

Description of the Diesel engine propelled ship, *Pacific*, and its power plant. The vessel made its trial runs last December and was at once taken over by the owners, who proposed to use it for the La Plate and Pacific Ocean business. It is 110 m. (360 feet) long over all, 15.5 m. (50.8 feet) beam, 7 m. (22.9 feet) draught and has a capacity of 6,500 tons. The engine room is very short and is located in the stern end of the ship. The ship has three decks and five loading hatches, each served by two electrically-driven winches.

The power plant consists of two directly reversible 4-stroke cycle Diesel engines having each six cylinders, 540 mm. (21.2 inches) bore and 730 mm. (28.7 inches) stroke, delivering together 2,000 I.H.P. Contrary to the practice of Diesel engine building in German shipyards, in this case the engines are enclosed and provided with lubrication under pressure. The arrangement, however, is such that the cylinders are left easily accessible. They are cast in blocks of three and, although they are open on the bottom, the piston rods pass through stuffing boxes, the purpose of which is to prevent dirt from the cylinders falling into the lubricating oil. The pistons are comparatively short and are equipped with self-tightening rings. Each cylinder cover has built into it a fuel valve, suction, exhaust, starting and safety valves, the starting valve being water cooled. Each engine has six fuel pumps—one for each cylinder—all driven from a common shaft. The pumps take the fuel by suction from two tanks located in the engine room, containing enough fuel for a run of 24 hours. They are filled every twelve hours, which gives the oil a chance to deposit some of the impurities which it contains.

In addition to the main engines there are two auxiliary Diesel engines, likewise 4-stroke cycle, each driving a dynamo and compressor, and each delivering 200 I.H.P. at 225 r.p.m. For the usual requirements of the ship one engine is sufficient, the other serving merely as an auxiliary unit. Each compressor delivers air at from 20 to 25 atmospheres. This air is stored in two steel tanks, from which it is taken to start the engines. Each of the main engines has in addition a high-pressure compressor which takes the air from the auxiliary compressor at 20 to 25 atmospheres and compresses it to 60 atmospheres. Each of the high-pressure compressors is large enough to deliver the entire air blast for the two main engines. All the pumps (two centrifugal pumps for the cooling water, two centrifugal pumps for the oil under pressure, two charging pumps, and two for drinking water) are driven separately by electric motors.

When loading and unloading the ship the compressors are uncoupled from the auxiliary Diesel engines which then serve to drive the dynamos

exclusively. One dynamo is in most cases sufficient to deliver all the current required for the winches and the other auxiliary machinery and lighting purposes. When the winches are not at work and the operation of a large Diesel engine would be economical, the lighting current is supplied by a small 110-volt dynamo, driven by an ordinary 2-stroke cycle crude-oil engine. This auxiliary dynamo also delivers current to drive a small high-pressure air compressor furnishing compressed air for starting the auxiliary Diesel engines.

The fuel carried for a return trip to South America is less than 700 tons; it can also carry about 1,000 tons of cargo. With an output of 2,032.5 I.H.P. the *Pacific* develops a speed of 11.41 knots.—"Page's Engineering Weekly."

BOILERS.

SUPERHEATED STEAM IN MARINE PRACTICE.

A paper, entitled as above, was read by Mr. Henry B. Oatley at the 23d General Meeting of the Society of Naval Architects and Marine Engineers, held in New York on November 18 and 19, 1915. The following is an abstract of this lengthy and important paper.

The present tendency with regard to superheating is to extend its application. This is true not only in marine practice, but also in locomotive and stationary engineering. It may be a matter of some surprise, to marine engineers generally, to note what is actually being accomplished with the higher degrees of superheat. The last decade has witnessed a wonderful advance in the adoption of high degrees of superheat on locomotives. About 40,000 locomotives throughout the world, of which more than 13,500 are in the United States and Canada, are today operating with temperatures in the steam chest above 600 degrees F. Some American railways are using 700 degrees temperature, while in France a large number of engines are operating daily and with wonderful success, at temperatures between 750 degrees and 800 degrees F. Many of these locomotives operate at piston speeds of 1,500 feet per minute. That these locomotives are not small engines is proved by the daily operation on several different railways of engines in both passenger and freight service which have shown above 3,000 I.H.P. We must not feel, therefore, that the temperatures used on locomotives are in combination with small engines, for the modern locomotive in the United States has practically the same indicated horsepower as the average cargo boat. In stationary practice the adoption of superheat has been very rapid, and a large number of plants in the United States are now in operation using at least 150 degrees of superheat. In marine practice during the last 15 years over 1,200 ships have been fitted with superheaters, all using 150 degrees of superheat, and the majority of which are operating with practically 200 degrees of superheat. It is not exaggeration to state that, at the present time, well over 2,000 steamers are afloat using superheated steam, the majority of which are operating with what we have classed as "high superheat." In reality the adoption of superheated steam is world wide, and is permeating nearly all branches of steam engineering.

Cylinder ratios to be suitable for use with highly superheated steam require consideration on account of the steam characteristics. The published dimensions of British and German superheated steamships show that a lower cylinder ratio is used. This is generally accomplished by increasing the diameter of the high-pressure cylinder, the intermediate and the low-pressure diameters, on a triple engine, remaining as before.

In quadruple engines, the high and first intermediate-pressure cylinders are increased in diameter, while no change is made in the second intermediate and low-pressure cylinders In converting saturated engines to the use of superheated steam an increase in the high-pressure cylinder diameter is not always advisable, and many times not possible. In such cases, if the indicated horsepower is to be obtained, it is necessary to increase the cut off in the high-pressure cylinder, thus giving an increased volume of steam at cut off and, in effect, decreasing the cylinder ratio. If the volume at cut off is not increased the indicated horsepower will be slightly decreased, assuming, of course, that the boiler pressure is not changed. To explain the necessity for these modifications it will only be necessary to point out that the expansion curve of superheated steam falls a little more rapidly than saturated steam and, to make up for this, the volume of steam at cut off, with, say, 200 degrees of superheat, should be about 10 per cent. greater than the volume with saturated steam in order to get equal power. This does not mean, of course, that the same weight of steam is used per stroke after superheating. It merely means that a 10 per cent. greater volume of steam, which weighs about 30 per cent. less per unit of volume than saturated steam, is used. These figures will indicate that there is still a considerable margin for steam economy, although a greater volume of steam per stroke is taken.

Lubrication is probably the most prominent point in mind when first considering the use of superheated steam. In reality this need not cause apprehension. With low degrees of superheat, it is not always necessary to use any oil internally. This may be explained by the fact that the superheat is not sufficient to carry the steam through the high-pressure cylinder without condensation. Where condensation occurs oil lubrication may not be necessary, and this is particularly true when low pressures are carried. Low superheat really does little more than eliminate part of the initial condensation. With moderate degrees of superheat opinions differ as to the necessity for the use of oil. Cases are on record where superheat of more than 100 degrees has been successfully used without internal oil lubrication. It seems probable, however, that in such cases heavy swabbing of rods was resorted to. When high degrees of superheat are considered, lubricating oil should certainly be applied internally. The quantity of oil needed to satisfactorily provide for 150 or more degrees of superheat is probably imagined as being greater than what is actually required. The average of a large number of boats using high degrees of superheat shows that very favorable results were obtained with about 0.3 gallons of oil per 1,000 H.P. per 24 hours. One case has been reported of a 2,800-H.P. ship that used but 20.5 gallons of oil on a 42-day voyage. This is just about 0.5 gallon per day, or 0.167 gallon per 1,000 H.P. per 24 hours. Mr. Gray, in his paper before the Institution of Naval Architects, speaking of the lubrication required, says:

"On the *Port Augusta*, of 2,000 I.H.P., 1.5 gallons per day of cylinder oil were used at first for internal lubricating and swabbing of piston rods of both main and auxiliary engines; on the *Port Lincoln* and *Port Macquarie*, of 4,000 I.H.P., about 2 gallons per day, this being gradually reduced until now only 0.5 to 0.75 gallon is required."

The keynote of the lubrication question with highly superheated steam is that a regular supply rather than a large quantity of oil is required. The oil furnishes the lubrication previously afforded by the water present in the saturated steam, but it does not follow that a regular supply of oil means a large quantity of oil.

An oil pump, driven from some convenient part of the engine, is required to feed oil in a steady and regular manner. Inexpensive pumps are on the market and are reported as being reliable in service. Many of them are so arranged that the oil container is not under pressure, thus permitting a renewal of the oil supply without trouble or danger. In

modern practice the oil is introduced into the steam pipe between the throttle valve and the engine. The delivery end of the oil-pipe fitting extends into the steam pipe, so that the current of steam entrains the oil and it is atomized before reaching the wearing parts. This feature has been found very important.

Filters are, of course, necessary where there is any possibility of oil being taken into the boilers. A large proportion of steamers using saturated steam in condensing engines are at present equipped with filters. It is believed that in a good many ships the filtering apparatus now installed, particularly with proper attention given to cleaning, is adequate to prevent oil reaching the boilers. A number of recently-built ships using high superheat have had two filters arranged in series, and reports from engineers on such boats indicate that oil does not get into the boilers so as to cause any damage or give trouble. No matter how good filters may be, they will not keep oil out of the boilers unless attended to. With a double filter installation, the first filter is generally inspected and cleaned about every two days while at sea. The second filter does not require attention oftener than about once a week.

Pyrometers are desirable for measuring the temperature of the steam at various points. It appears, at the present time, common practice to have one pyrometer indicating the temperature of steam at the engine stop valve, also to have other pyrometers at points nearer the superheater for indicating the temperature of the steam where it leaves the superheating apparatus. A pyrometer to register the temperature of steam in the intermediate-pressure chest is also frequently applied. Various types of pyrometers which measure these temperatures accurately are available and are reliable instruments; the electric pyrometer, especially, appears to have been developed in a satisfactory manner for marine practice. The information given by the pyrometer is important, and makes it possible to determine:

1. How far superheat is being carried through the engine.
2. Whether any boiler is being properly or improperly fired.
3. By showing a reduction in steam temperature, gives warning of an impending fall in boiler pressure; in other words, acts as a danger signal as regards pressures.—"Shipbuilding and Shipping Record."

THE "DEAN" BOILER-TUBE CLEANER.

The "Dean" Boiler-Tube Cleaner possesses the unique feature of being adaptable for use in either water tube or return tubular boilers. By simply changing the vibrator head or hammer the cleaner becomes equally efficient in removing scale from the interior or exterior of boiler tubes. The device is purely vibratory in its action, being therefore entirely free from any boring, scraping or other cutting or frictional action.

A sectional view of the apparatus is shown in Fig. 1, from which it will be seen that the vibrating head might be likened to a power-operated pendulum, and its effect on the walls of the tube is to create a state of localized vibration or tremor such as is caused in a bell when struck by the clapper. Owing to the difference in hardness and strength between hard scale and steel boiler tubes, the scale is physically unable to vibrate at the high speed of the steel. Consequently, the surfaces in contact are separated from each other almost immediately, and a continuance of the vibration produces an internal disintegration of the scale causing it to fall away from the tube surfaces in small pieces, which, in the case of

water-tube boilers, are expelled from the tube by the exhaust air or steam which operates the cleaner.

This peculiar action renders the "Dean" tube cleaner particularly efficient in removing scale from corners, crevices, or joints which are difficult of access.

Referring to Fig. 1, the vibrator (a) is hinged at (b), and carries at its outer end the vibrator head (c), while the other end is formed so as to operate in valve (d). The hinge (b) is carried by body (e) in which are formed passages (f), one end of which is controlled by valve (d), the other ends entering cylinder (g) at opposite ends. Piston (h) surrounds vibrator (a), a ball-and-socket arrangement being provided so that vibrator (a) may be moved by piston (h).

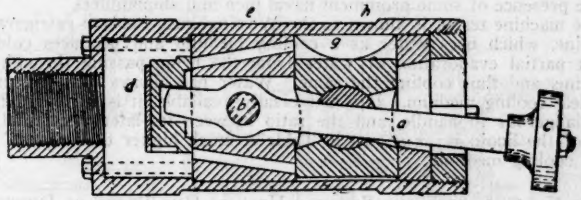


FIG. 1.

With the vibrator head in the lever position as shown, the valve admits the working fluid to the lower end of the cylinder, whereupon the piston is instantly forced upward, carrying the vibrator with it, and causing the head to make impact with the upper side of the tube. The action of the vibrator causes the valve to travel downward and uncover the upper passage to the cylinder, thus returning the piston to the position shown.

The tapping action takes place at the rate of from 3,500 to 10,000 times per minute and gives rise to the series of vibrations in the boiler tubes which result in the scale automatically detaching itself from the surface. Again, the vibratory movement can be regulated to such a mechanical nicety that at the highest possible speed the tube is not subjected to hammering in the usual sense of the word—a gentle tapping of very high frequency being what actually takes place.

The actual force of the tap is equivalent to 2½ foot ounces, and with 70 to 90 pounds pressure it vibrates at a speed of 5,000 to 8,000 times per minute. The Dean cleaner is manufactured by the William B. Pierce Co., of Buffalo, New York.—"Page's Engineering Weekly."

A STUDY OF BOILER LOSSES, by A. P. KRATZ, Assistant in Mechanical Engineering, University of Illinois.

Results of numerous tests made to permit of a detailed study of the boiler and furnace losses under varying conditions of load, depth of fuel bed, and draft.—"Power," Vol. 42, No. 22, November 30, 1915.

SUDDEN COOLING OF BOILER IN CASES OF LOW WATER, by R. H. BLACKBURN, Chief Inspector of Steam Boilers, Saskatchewan, Canada. A defense of the statement that the sudden cooling of overheated boiler plates is the safest practice in cases of low water.—"Power," Vol. 42, No. 20, November 16, 1915.

AUXILIARIES.

—

NEW MARINE REFRIGERATING MACHINE.

DESCRIPTION OF CONSTRUCTION AND OPERATION OF WESTINGHOUSE-LEBLANC REFRIGERATING MACHINE DESIGNED FOR MARINE WORK.

By G. L. Kothny.*

On Friday, October 29, 1915, a public demonstration of a new type of refrigerating machine especially suitable for marine installations was given at the Westinghouse Machine Company's works at East Pittsburgh, Pa., in the presence of some prominent naval men and shipbuilders.

The machine tested is known as the Westinghouse-Leblanc refrigerating machine, which uses water as a cooling medium and produces cold by direct partial evaporation of water from the brine passing through the machine, and thus cooling the same. Water has always been known as an ideal cooling medium. It is universally available, it is neither difficult nor dangerous to handle, and the ratio between its latent heat and the heat of the liquid is, as shown in Table 1, much higher than that of any other cooling medium.

Table No. 1.—Latent and Sensible Heat of One Pound of Different Liquids for 14 Degrees F. Liquid Temperature and 68 Degrees F. Condenser Temperature.

	Ammonia NH_3.	Carbon Dioxide CO_2.	Sulphur Dioxide SO_2.	Water H_2O.
Pressure in brine cooler, lbs. per sq. in.....	41.5	385.0	14.6	0.0398
Condenser pressure, lbs. per sq. in.	125.0	825.0	76.0	0.3270
Latent heat, in B.t.u....	558.0	111.0	172.0	1050.0000
Heat of liquid, in B.t.u.	51.0	32.0	17.4	54.0000
Heat units available for producing cold, per lb.	507.0	79.0	155.6	996.0000
Per cent., taking water as a base	51.0	7.9	15.7	100.0000

The evaporation of water in the Westinghouse-Leblanc machine is obtained by producing and maintaining a vacuum, which corresponds to the vapor tension of the water at the desired brine temperature, in an evaporator through which the brine circulates. In other words, if, for instance, it is desired to produce brine at 14 degrees F. temperature, a vacuum corresponding to the vapor tension of water at 14 degrees F. (which is 0.08 inch Hg. absolute), or of 29.92 inch Hg. with 30-inch barometer, has to be produced and maintained in the evaporator, the vessel through which the brine circulates and in which it is cooled.

In the Westinghouse-Leblanc machine this is accomplished by a combination of steam ejector or ejectors and condensing plant, the ejector or ejectors producing the difference in vacuum between that necessary in the evaporator and the highest possible vacuum which may be obtained in the condenser with circulating water of a certain given temperature. A diagrammatic arrangement is shown in Fig. 1. The operation is as follows:

*Engineer, the Westinghouse Machine Company, East Pittsburgh, Pa.

Live steam enters the steam turbine (*A*) after passing through a moisture evaporator with a steam pressure of about 200 pounds gage, and develops the necessary energy in the turbine for driving the pumps, which are coupled directly to the turbine. The steam leaves the turbine with a back pressure of about 55 pounds and passes through the automatic shut-off valve *B* and the steam globe valves *C-C* to the steam ejectors (marked first and second ejector). The object of the automatic shut-off valve is to prevent steam from entering the ejectors as long as there is no vacuum in the evaporator or condenser. The valve itself is controlled by vacuum and will not operate unless a designed vacuum exists in the condenser.

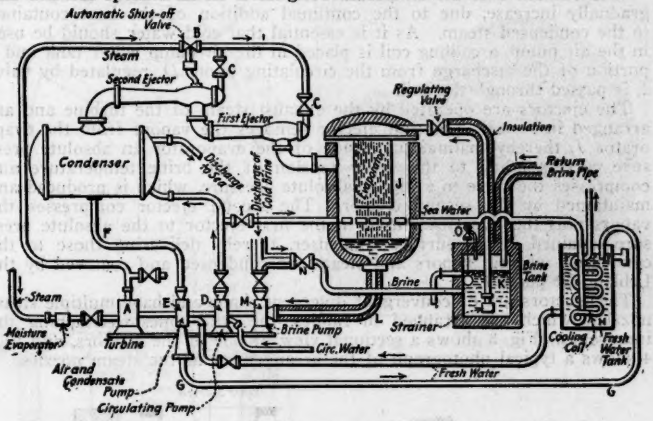


FIG. 1.—DIAGRAMMATIC ARRANGEMENT OF WESTINGHOUSE-LEBLANC REFRIGERATING MACHINE.

The high vacuum in the evaporator is produced and maintained by the two steam ejectors, working in conjunction with a surface condensing plant. The surface condensing plant consists of a surface condenser, condenser circulating pump (*D*) and Leblanc air and condensate pump (*E*). Circulating water enters the circulating pump, passes through the pump, which is of the centrifugal type, and also through a non-return valve, then through a pipe as indicated by the arrows into the condenser, and is finally discharged back to the sea. The air and condensed steam are taken out from the condenser through pipe *F* and removed by the air pump *E*, which is of the rotary hydraulic type, through pipe *G* into the fresh-water tank *H*.

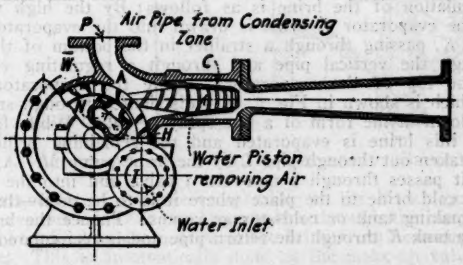


FIG. 2.—SECTION OF WESTINGHOUSE-LEBLANC AIR PUMP.

A cross-sectional view of the Westinghouse-Leblanc air pump is shown in Fig. 2. The principle of operation is illustrated in this figure. Water enters the chamber *H* through the opening *I* and flows out through the orifice *N*. The impeller *W*, rotating in a clockwise direction, cuts off layers of water and projects them into the cone *C*. Between the successive pistons of water, layers of air drawn in through opening *P* are imprisoned. The high velocity of these water pistons is transformed into pressure by means of the diffusing cone so the moisture may be discharged against atmospheric or somewhat higher pressure.

It is obvious that the temperature of water in tank *H* (Fig. 1) would gradually increase, due to the continual addition of the heat contained in the condensed steam. As it is essential that cold water should be used in the air pump, a cooling coil is placed in the air-pump water tank and a portion of the discharge from the circulating pump *D*, regulated by valve *I*, is passed through the coil.

The ejectors are operated by the exhaust steam of the turbine and are arranged in series. The first ejector removes the vapors from the evaporator *J*, thereby maintaining inside of the evaporator an absolute pressure corresponding to the vapor tension of the brine temperature and compresses the same to a higher absolute pressure, which is produced and maintained by the second ejector. The second ejector compresses the vapors plus the live steam used in the first ejector to the absolute pressure obtained in the surface condenser, thereby delivering those to the condenser, wherein vapors and steam are condensed and removed by the Leblanc air pump *E*.

The ejectors are of convergent divergent shape and have multiple steam nozzles, which are arranged in ranges, the outer ones overlapping the inner ones. Fig. 3 shows a sectional view of one of the ejectors, and Fig. 4 shows a typical photograph of the arrangement of the steam nozzles.

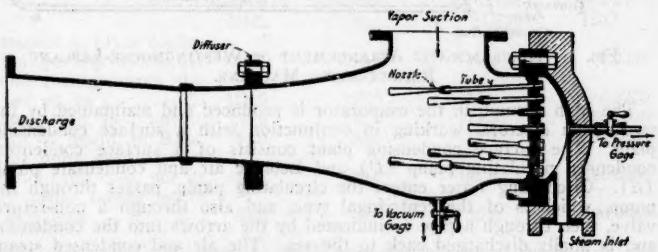


FIG. 3.—SECTION OF WESTINGHOUSE-LEBLANC STEAM EJECTOR.

The circulation of the brine is as follows: By the high vacuum existing in the evaporator *J*, brine is drawn into the evaporator from the brine tank *K*, passing through a strainer in the bottom of the tank and then through the vertical pipe and through a regulating valve before entering the top of the evaporator. Inside the evaporator a section through which is shown in Fig. 5, it is distributed through small orifices and falls down in the form of a fine spray or rain. While falling down a part of this brine is evaporated and the remaining brine is cooled down and taken out through pipe *L* by the brine pump *M*. After leaving the pump it passes through a non-return valve and into the pipe which carries the cold brine to the place where it is to be used—that is, either to the ice-making tank or cold-storage rooms. Thence the brine returns to the brine tank *K* through the return pipe and is recirculated as already described.

An automatic brine pressure-relief valve *N* is provided at the discharge of the brine pump, and the outlet of this valve is connected directly to the brine tank *K*. The object of this valve is to allow the brine to return automatically to the brine tank in case all the valves in the brine system (which may be located far away from the machine) are accidentally closed and thus prevent the discharge of brine from the pump, which would result in the filling up of the evaporator. Fig. 6 shows a section through this valve which may be adjusted to operate at any specified pressure.

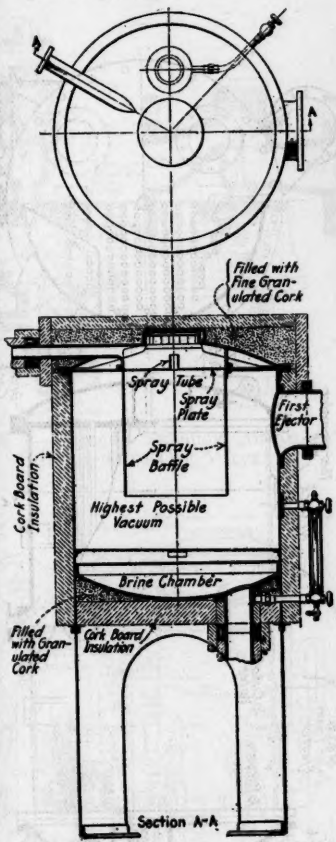


FIG. 5.—EVAPORATOR.

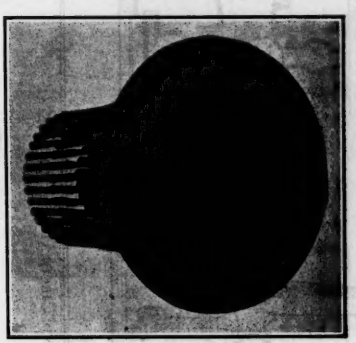


FIG. 4.—EJECTOR NOZZLES.

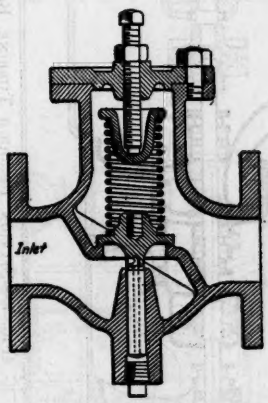


FIG. 6.—SECTION OF BRINE RELIEF VALVE.

A certain amount of water corresponding to the B.t.u.'s output of the machine is evaporated while the brine passes through the evaporator, and this has to be added again to prevent the brine from becoming more and more dense. This is automatically done by the make-up valve *O*, which

opens whenever the brine level in the brine tank lowers and allows some sea water to flow into the brine tank until the normal brine level is reached.

A section through the driving steam turbine, the circulating, air and brine pumps is shown in Fig. 7. The turbine is of the single-wheel impulse type and has an automatic stop governor which prevents the turbine

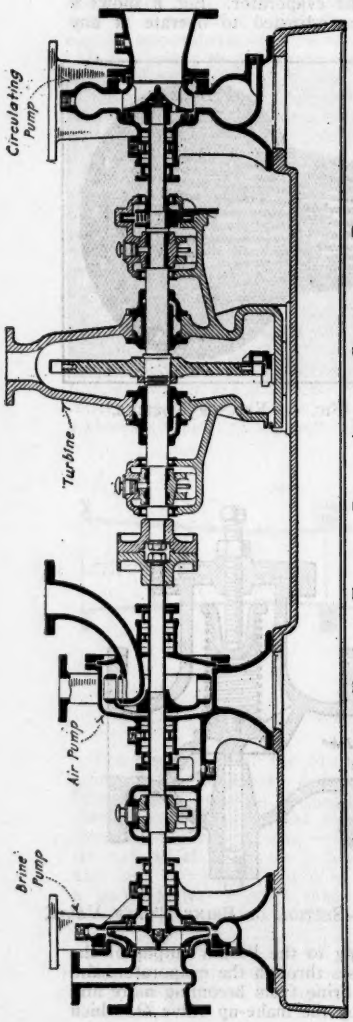


FIG. 7.—SECTION THROUGH TURBINE, BRINE, AIR AND CIRCULATING PUMPS.

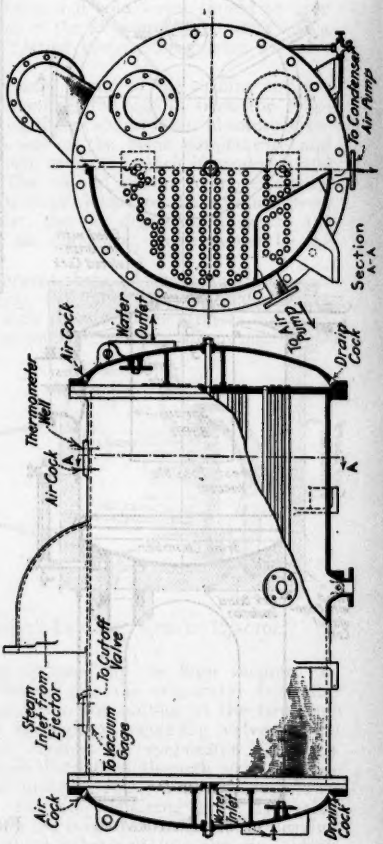


FIG. 8.—CONDENSER.

and pumps from running at an undesirable speed. As all of the pumps are of the centrifugal type, and two of them are always submerged, there is no necessity to provide a speed-regulating governor. All the pumps have automatically water or brine-sealed glands which insure their being airtight.

Fig. 8 shows a section through the surface condenser. The condenser is of the four-pass type and has 5/8-inch O. D. tubes, No. 18 B.W.G.; the air suction is arranged on the side of the condenser, while the condensate outlet is located at the bottom.

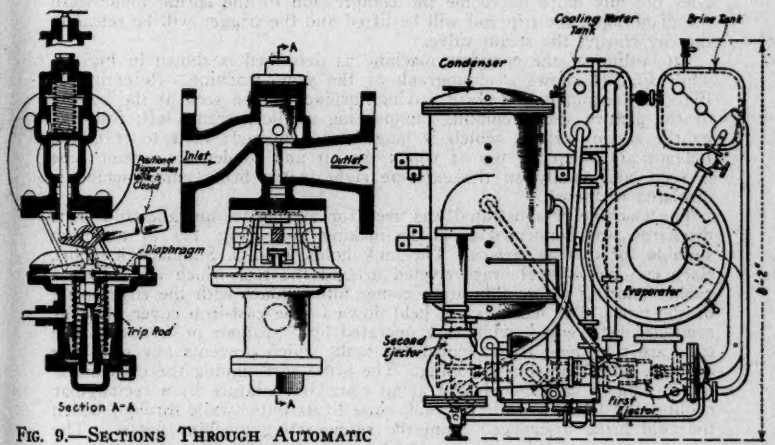


FIG. 9.—SECTIONS THROUGH AUTOMATIC STEAM SHUT-OFF VALVE.

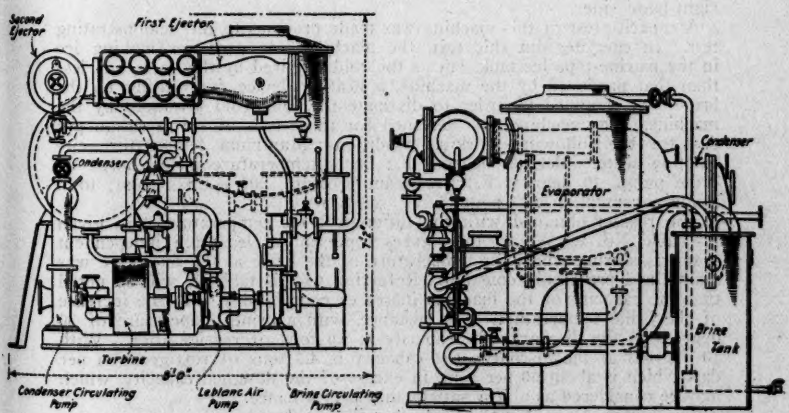


FIG. 10.—OUTLINE OF REFRIGERATING MACHINE, SHOWING PLAN AND FRONT AND SIDE ELEVATIONS.

Fig. 9 shows the arrangement and also illustrates the operation of the automatic shut-off valve *B* (Fig. 1.). A flexible diaphragm reinforced by a steel plate and held upwards by a spring forms the top cover of a small cylinder which is connected to the surface condenser. When the vacuum reaches a certain point, the atmospheric pressure will overcome the compression of the spring and move the trip rod downwards. If the trigger operating the steam valve is now pressed down, the catcher on the trigger will be caught by the trip rod and the valve will be held open. As soon as the vacuum drops to a point where the atmospheric pressure does not any more overcome the compression of the spring underneath the diaphragm, the trip rod will be lifted and the trigger will be released, thereby closing the steam valve.

An outline of the complete machine as described is shown in Fig. 10, while Fig. 11 shows a photograph of the same machine. Referring to Fig. 11, the pumps and their driving turbine will be seen at the bottom of the picture, the circulating pump being at the extreme left, followed by the steam turbine, which is lagged and covered; next to it is the Leblanc air pump, on top of which the air and condensate suction pipe is seen, and finally on the extreme right is the brine pump, which is insulated with cork.

The machine demonstrated was used for making ice and the cold brine discharge was connected to an ice-making tank (Fig. 12) of a design suitable for marine service. The tank holds 9 U. S. Standard ice cans. Each can has a steel brace riveted around the top, which also holds a rubber packing where this brace comes into contact with the cover plate of the tank. The ice cans are held down to the cast-iron cover plate by semi-circular wedges, which are operated by a spanner or wrench. The cans are thus held tight against the tank, which prevents any splashing of brine when the ship is rolling. The brine surrounding the cans inside of the tank is cooled and kept at an equal temperature by a rectangular cooling coil located inside the tank close to its outer walls through which the cold brine discharged from the refrigerating machine passes. The ice cans are covered with wooden lids, which prevent any splashing of fresh water when the cans are filled up at the beginning of the process of freezing. The circulating brine inlet and outlet will be seen on the right-hand side.

A capacity test of this machine was made previous to the demonstrating test. In carrying out this test, the machine was used for making ice in the marine-type ice tank, but as the cold required by this tank was less than that produced by the machine, a heating device for heating up the brine was arranged in order to dissipate the total cold produced by the machine. The machine was designed for three tons of refrigeration per day for the following working conditions: Maximum temperature circulating water inlet, 85 degrees F.; brine temperature at discharge of brine pump, 10 degrees F.; live-steam pressure, 200 pounds gage; total steam consumption, 955 pounds.

The results obtained with the different brine temperatures are given in Table No. 2. All of the figures represent at least ten independent readings, during which the temperature of the brine at the discharge was maintained practically constant. Referring to this table, it will be noted that the capacity of the machine increases considerably with the increase of the brine temperature. For instance, with a brine temperature of 30 degrees at the discharge the capacity is 8.6 tons of refrigeration. With 10 degrees at the discharge the capacity is 4.5 tons of refrigeration per day, which is about 50 per cent. in excess of the designed capacity, which may be considered as a very satisfactory performance.

For higher brine temperatures it will not be necessary to use two ejectors in series. One ejector will do the compression work, and naturally the steam consumption of this ejector will become considerably less

TABLE NO. 2.—TEST RESULTS (WESTINGHOUSE-LEBLANC) OF NAVY-TYPE REFRIGERATING MACHINE TESTED AUG. 4 AND 5, 1915. LIVE STEAM, 200 LBS./IN². BAROMETER 29.17 INCHES HG. TEMPERATURE OF ATMOSPHERE, 85 DEGREES TO 87 DEGREES F.

Vacuum, inches of mercury.			Circulating water temperature.		Brine.						Refrigeration.				Steam consumption.		
					Temperature.			Quantity.									
First ejector.	Second ejector.	Condenser.	Inlet to condenser.	Within fresh-water tank.	Inlet.	Discharge.	Difference.	Pounds per hour.	Density salometer.	Spec. heat.	B.t.u. per hour.	Tons per day.	Loss through frosted surface, tons per day.	Total tons.	Steam and vapor per hour.	Vapor per hour.	Total steam consumed by turbine and ejectors.
28.95	28.95	27.35	86.0	95	35.50	30.05	5.45	19,080	76	.834	99,598	8.30	0.32	8.62	1,276	106	1,170
28.95	28.95	27.35	85.5	94	27.00	22.80	4.20	19,080	80	.829	76,702	6.39	0.35	6.74	1,276	82	1,194
29.00	28.95	27.80	82.5	92	18.40	15.00	3.40	17,240	77	.833	56,202	4.68	0.37	5.05	1,125	61	1,064
29.00	28.95	27.80	83.0	92	13.00	10.00	3.00	17,240	80	.829	49,479	4.13	0.42	4.55	1,102	53	1,049
29.00	28.95	27.60	86.0	94	13.15	10.10	3.05	17,240	80	.829	50,341	4.20	0.42	4.62	1,042	54	988
29.00	28.95	27.60	84.0	93	11.85	8.90	2.95	16,800	80	.829	47,544	3.96	0.45	4.41	975	52	923
29.00	28.95	27.60	84.5	94	12.80	9.80	3.00	17,240	80	.829	49,479	4.13	0.42	4.55	1,013	53	960

The loss of refrigeration through frosted surfaces is based on 21 ft.² actually measured. Above readings are the averages of 10 independent readings for each temperature range, taken at intervals of 10 minutes.

than that of two ejectors. The pumps may also be motor driven if more convenient.

The advantages of this refrigerating machine may be summarized as follows:

The Complete Absence of Chemicals or Compressed Gases.—This is considered of great importance for navy work, as it makes ships using this type of machine naturally independent of the supply of chemicals. They do not need to carry a supply of flasks containing gases at a high pressure, which are a source of danger to the vessel itself, nor will the lack of chemicals interfere with the refrigeration. They can produce refrigeration as long as there is coal available on board to make steam. For a warship this is a most important consideration. In times of war it might be impossible for the vessel to obtain a supply of chemicals, making it necessary to discontinue refrigeration, thereby spoiling the foodstuff in tropical or semi-tropical climates. The powder and ammunition magazines may also be endangered, as some powders or explosives, unless properly cooled, deteriorate and explode from self-combustion. Furthermore, flasks containing gas under pressures of 600 to 1,000 pounds per square inch, if struck by one of the enemy's projectiles, would cause as much damage to the ship as the explosion of the projectile itself. In the machine described above no gases are used and no pressures have to be maintained. The highest pressure is atmospheric pressure and the cooling medium used are water vapors.

Minimum Weight and Space Occupied.—This is mostly due to the absence of high pressures inside the machine and also to the direct cooling of the brine. The construction can be very light and at the same time rigid enough to stand atmospheric pressure. It is claimed that the weight of this machine and the space occupied by it are only one-third or one-half that of other types.

Simple Operation.—All of the pumps are of the centrifugal-pump type, and form a single revolving element without close clearances. There are no valves to be kept tight, no piston rods to pack and no parts to adjust. The operation is practically automatic and requires no specially trained engineers. The running is silent, a point of particular importance where the apparatus is used on passenger steamers.

Low Cost of Operation and Maintenance.—The reason for this is obvious, when the simplicity of the apparatus is considered. Practically no repairs are needed, there are no chemicals and no high pressures to handle.

Efficiency.—Tests have shown that the results compare very favorably with those of other types, especially when the inlet temperature of the circulating water is around 85 degrees F.

Brine Coolers Eliminated.—Most of the other types of machines use a brine cooler, which forms a considerable item in weight and space on board the ship. In this machine the brine is cooled directly in the evaporator, and therefore does not require any separate brine cooler.

These many advantages have induced the navies and many of the steamship companies of France, Russia, Austria, Germany, England, Japan, Spain, Greece and Argentine to use this machine on their vessels for ice making, cold storage and magazine cooling. Up to June, 1914, about 250 machines had been installed and ordered in these different countries for marine work, some of them having been in service for over three years, giving excellent results without interruption in service or ever being subjected to any repairs.

A photograph of the type of machine used in the French Navy is shown in Fig. 13. The machine illustrated is of 8 tons refrigeration capacity and is motor driven. Fig. 14 shows a 10-ton turbine-driven machine. Several of these have been installed on the following steamers: *André Lebon, Athos, Porthos, Sphinx*, of the Compagnie des Messayeries Maritinies, Marseilles, France. These steamers ply between Marseilles, France,

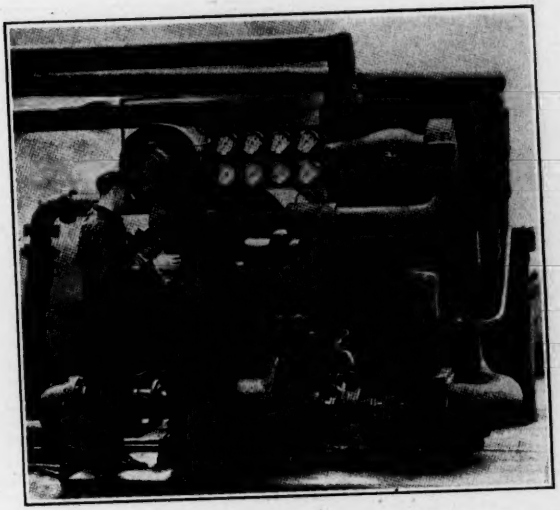


FIG. 11.—REFRIGERATING MACHINE UNDER TEST AT WESTINGHOUSE WORKS.

FIG. 12.—MARINE TYPE ICE-MAKING TANK.

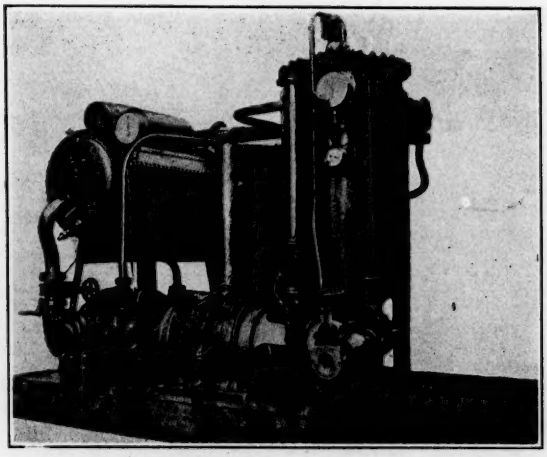


FIG. 13.—MOTOR-DRIVEN REFRIGERATING MACHINE USED IN THE FRENCH NAVY.

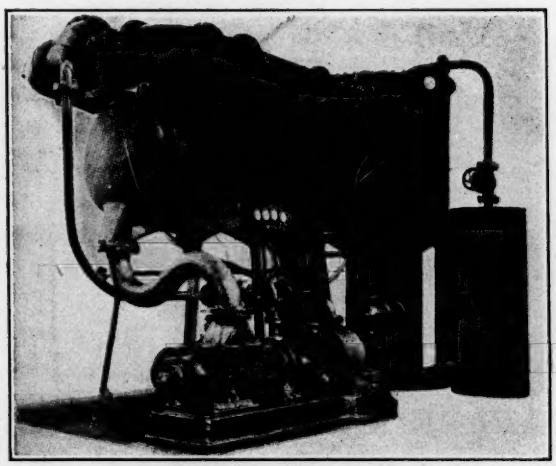


FIG. 14.—TURBINE-DRIVEN REFRIGERATING MACHINE ON BOARD S. S. "ANDRE LEBON."

and Yokohama, Japan, passing through the Red Sea, the Indian Ocean and the South China Sea. In those waters the temperature of the circulating water often reaches 90 degrees F., and in spite of this severe condition the machines installed have successfully met all specifications under test.

The Westinghouse Machine Company, of East Pittsburgh, Pa., who are the sole licensees of the Leblanc patents in the United States, have recently taken up the manufacture of this machine for marine as well as for land installations.—"International Marine Engineering."

NEW VACUUM APPARATUS.

WESTINGHOUSE-LEBLANC AIR-EJECTOR SYSTEM FOR PRODUCING HIGH VACUUM.

By G. L. Kothny.*

On the occasion of the demonstration of tests of a new type of refrigerating machine at the Westinghouse Machine Company's works at East Pittsburgh, Pa., which is described on another page in this issue, a new high-vacuum producing device was also shown to the visitors.

This device is called the Westinghouse-Leblanc air-ejector system. It is a substitute for the ordinary reciprocating vacuum pump, or any other vacuum pump of hydraulic or rotary-piston type, and is designed to produce a very high vacuum in the condenser. The air and non-condensable gases are taken out of the condenser by an air ejector of special design operated by live steam, and the condensed steam is removed from the condenser by a turbine-driven condensate pump of the centrifugal type. A diagrammatic arrangement of this Westinghouse-Leblanc air-ejector system is shown in Fig. 1.

Referring to this figure, the operation is as follows: Live steam of any pressure available from the boilers enters the ejector, which acts as a dry-vacuum pump, and withdraws the air from the condenser, thereby creating a high vacuum in the same. The condensate, which collects at the bottom of the main condenser, is taken out by the condensate pump and discharged into an air-separating tank, where the steam and vapor coming from the air ejector are condensed by the condensate discharged from the condensate pump. The temperature of the condensate is naturally increased by the heat given to it from the steam discharged from the air ejector. The non-condensable gases and the air removed from the condenser are separated in this tank and escape to the atmosphere through a vent pipe which is arranged on top of the tank. The condensate flows by gravity to the feed and filter tank, from which it is taken by the boiler-feed pumps into the boilers. As will easily be seen, the apparatus has the highest thermal efficiency possible, as all the heat supplied is regained in the feed water.

The condensate pump is of the centrifugal type and of special design to operate with a low submergency on the suction side, in order to avoid a high location of the condenser. It is driven by a small steam turbine, the shaft of the turbine being extended to bear the impeller of the pump, which has a single inlet, thus requiring only one gland. This gland is automatically water sealed, thus insuring airtightness. Fig. 2 shows such a pump, having a capacity of 130,000 pounds of steam per hour when working with 12-inch submergency against a total head of 65 feet. It will be noted that the size of the pump is very small, considering its working conditions. The weight of the complete pump, turbine and bed plate is 750 pounds.

*Engineer, the Westinghouse Machine Company, East Pittsburgh, Pa.

The air ejector is shown in Fig. 3. Two ejectors as illustrated are sufficient for a condensing plant condensing 100,000 pounds of steam per hour, when maintaining a vacuum of 28½ inches with 60 degrees of circulating water. The length of the ejector shown is 3 feet and its weight is 97 pounds.

Referring back to Fig. 1, a circulating pipe will be seen connecting the main condenser with the feed-water tank. The purpose of this pipe is explained by the following: It might occur sometimes that the main engines are stopped and that very little steam is entering the main condenser from the auxiliaries, so that there is a very small quantity of condensate to be removed from the condenser. If at the same time the vacuum is to be maintained in the condenser the steam coming from the air ejector would have very little water to be condensed with, and the continuous adding of heat by the steam from the ejector may heat up the feed water to an undesirably high temperature. To avoid this and to enable a desired temperature to be maintained in the feed tank, the circulating pipe is provided.

By opening the valve shown in this pipe, hot water from the feed tank is sucked into the condenser by the vacuum, and, while falling over the cold condenser tubes, it is cooled. The cooled water is taken out by the condensate pump and discharged into the air-separating tank, where it condenses the steam coming from the air ejector and from where it flows to the feed tank. Part of this water is returned to the condenser and recirculated as described. By regulating the quantity of water returned to the condenser, the temperature of the feed water in the feed tank may be regulated. If, however, a number of auxiliaries are operated while the main engine is idle, the exhaust from the auxiliaries may enter the main condenser and sufficient water will be taken out by the condensate pump for condensing the steam coming from the air ejectors.

Instead of using one ejector, two or three of a smaller size may be arranged in parallel for one condenser, thus allowing one or two ejectors to be shut down (thereby saving one-half or one-third of the steam consumption) when running at low speed or during the cold season, when the vacuum is higher on account of the low temperature of the circulating water. Fig. 4 shows a diagrammatic arrangement of two ejectors in parallel. Gate valves have to be provided at the air suction. The gate valve on the ejector which is not in operation should be closed in order to prevent the atmospheric air from passing through this ejector back into the condenser.

Fig. 5 shows an air-ejector installation at the Westinghouse Machine Company's works at East Pittsburgh, Pa. At the bottom of this picture and to the extreme left the turbine-driven condensate pump is shown. Next to it is the air-separating tank, and at the right-hand side the air ejector.

An installation of this kind has also been made on the United States torpedo boat tender *Melville*, built by the New York Shipbuilding Company, Camden, N. J. The air-ejector system was installed in parallel with the already existing reciprocating air pump and vacuum augmentor, in order to enable a true comparison of both systems under absolutely identical conditions. The performance of the air-ejector system on the *Melville* has been very successful; the vacuum obtained was higher, with less steam consumption than that obtained by the augmentor. A higher feed-water temperature was also obtained and the weight and space occupied by it are considerably less than that of the augmentor system.

The advantages of this new device may be summarized as follows:

Minimum Weight.—A few figures will illustrate this. Taking for example the installation of the main air pump for one main condenser on a battleship: The size of such a pump is 18 inches by 36 inches by 36 inches by 27 inches; its weight about 15,000 pounds. To this would have to be

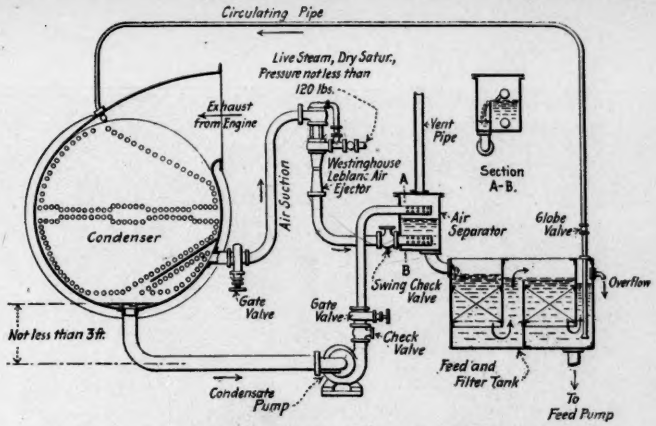


FIG. 1.—DIAGRAMMATIC ARRANGEMENT OF WESTINGHOUSE–LEBLANC AIR-EJECTOR SYSTEM.

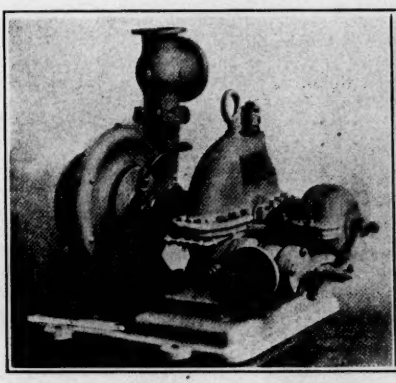


FIG. 2.—CONDENSATE PUMP.

FIG. 3.—AIR EJECTOR.

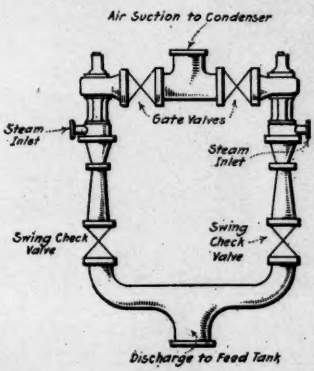


FIG. 4.—ARRANGEMENT OF TWO AIR EJECTORS IN PARALLEL.

FIG. 5.—AIR EJECTOR INSTALLATION AT THE WESTINGHOUSE MACHINE COMPANY'S WORKS.

added the weight of the augmentor, the augmentor condenser, the water seal and the inter-connecting piping between the different parts of the augmentor installation, which is about 5,500 pounds, the total weight of one complete air-pump installation with augmentor being 20,500 pounds without water.

The Westinghouse-Leblanc air-ejector installation would weigh as follows:

Air ejectors	650	pounds
Condensate pump, turbine and bed-plate	1,000	"
Valves, accessories and air-separating tank	920	"
Total weight of installation	2,570	pounds

against 20,500 pounds of the reciprocating-pump installation, or the air-ejector installation weighs only about one-eighth of the reciprocating-pump installation. This does not include the difference in weight of the piping, the sizes of which are considerably larger for the reciprocating-pump installation, and when filled with water will be considerably greater in weight.

Minimum Space.—The space occupied by the Leblanc air-ejector system is from one-eighth to one-tenth that of the reciprocating-pump installation. Furthermore, the air ejector has the advantage that it may be located anywhere, just like a piece of pipe, and also does not need to be in a certain relative position to the main condenser or to the condensate pump. The reciprocating pump and the vacuum augmentor, augmentor condenser and water seal, however, must always be arranged in a relative position to each other, which makes the general arrangement of the augmentor installation difficult and complicated.

High Thermal Efficiency.—All steam used in the air ejector and the turbine driving the condensate pump is condensed by the condensed steam from the main condenser. The temperature of the cendensate is raised by the heat contained in the steam of the air ejectors and the turbine, while in the augmentor system the heat contained in the steam operating the augmentor is thrown overboard as it is transmitted to the circulating water passing through the augmentor condenser.

Simplicity.—The Leblanc air ejector is a static apparatus, and therefore does not require any attention during its operation. The condensate pump is of the centrifugal type without close clearances or valves to look after, and all that needs attention is the turbine bearings, since they should be supplied with sufficient lubricating oil. However, if there is a feed-oil system on board the ship, it may be used for supplying oil to these bearings, and then the condensate pump will not require any attention.

Maintenance.—The complete absence of internal rubbing parts, of reciprocating valves, etc., eliminates the wear and tear, and no repairs are required, which are frequent with pumps of the reciprocating type.

Small Steam Consumption.—The steam consumption of the air-ejector system is less than that of a reciprocating pump working in conjunction with a vacuum augmentor, when working under similar conditions.

The Westinghouse Machine Company, of East Pittsburgh, Pa., the sole licensee for all Leblanc patents relating to condensing apparatus, is now manufacturing this air ejector for marine condenser installations.—"International Marine Engineering."

FIG. 5.—AIR EJECTOR INSTALLATION AT THE WESTINGHOUSE MACHINE COMPANY'S WORKS.

THE APPLICATION OF SMALL STEAM TURBINES FOR AUXILIARY PURPOSES ON BOARD SHIP.—Abstract of a paper read by MR. W. J. A. LONDON and MR. FREDERICK D. HERBERT at the 23d General Meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 18 and 19.—"Shipbuilding and Shipping Record," December 2, 1915.

MODERN REFRIGERATING PLANT AND ITS USES, by W. S. DOUGLAS, B. Sc.—Treats of ordinary refrigerating plants of various types, recent developments in machines using water vapor, the application of multi-effect compression and new types of machines; the application of machines, and the advance in the theory governing their construction.

OIL AND OIL ENGINES.

McINTOSH & SEYMOUR 500-H.P. DIESEL ENGINE.

BY F. R. LOW.

SYNOPSIS.—Description of 500-H.P. vertical, 4-cylinder engine built by the McIntosh & Seymour Corporation under license from the Swedish Diesel Engine Co.

The idea is prevalent that the Diesel engine is a complicated piece of machinery, but when one looks at the reproduced photographs he sees all that there is of a Diesel-engine plant; whereas a steam engine must have a boiler, heater, feed pump, injector, condenser and an air pump. The man that stands upon the platform of a Diesel engine has the whole plant under his eye. Furthermore, the engine illustrated is really four engines, all alike and each extremely simple—a shaft, connecting rod, piston, inlet valve, exhaust valve and a fuel valve, with mechanism for operating and controlling the valves, constitute the entire list of working part for each one of these engines.

In Fig. 4 *A* is the top of the air-inlet valve cage. The valve opens inwardly and admits air, which is drawn through the slotted intake *B* as the piston makes its downward stroke. *C* is the top of the exhaust-valve cage. This valve also opens inwardly and is held open by the hooked lever *D* when the engine is turned over by hand. The exhaust valve itself and the cage detached are shown in Fig. 11, where *A* is the renewable seat on the valve and *B* the water jacket for the stem. The valve is assembled by slipping the cage over the stem, putting the spring in place, compressing it until the slotted collar *C* can be slipped into the groove *D* and allowed to settle into the pocket *E* in the top of the spring cap. The air-admission valve is similar to the exhaust valve except that its stem guide, being in contact only with cool air, is not water-jacketed.

The exhaust valve is the part of a Diesel engine requiring the most frequent attention, since the hot gases impinge on it more in exhausting than on any other. Under the most unfavorable conditions, however, it ought not to be necessary to inspect, grind and clean this valve more than once in every six weeks. The process is very simple. The shaft *F*, Fig. 4, which carries the levers that operate the valves, has grooves near the ends, best seen in Fig. 9. The nuts *GG*, Figs. 4 and 9, are removed, the housing of the fuel valve unshipped and a sling, taking hold of the shaft at the grooved ends, lifts it free with its system of levers. Then, by removing the nuts *HH*, Fig. 4, and unmaking the jacket-water connections, the exhaust valve can be lifted out, a spare inserted and the engine put into running condition again in a few minutes; the valve taken out being cleaned and put in order at one's leisure. The air and fuel valves require only infrequent attention, and a regular schedule of a valve once in a fortnight will keep a 4-cylinder engine in excellent condition, under ordinary circumstances. This, with the inspection and cleaning of each cylinder once in every 6 to 9 months, together with ordinary attention to the bearings, is all that the engine requires. This certainly compares favorably with boiler cleaning and furnace repairs, the packing of stuffing boxes and the leaky pipe connections requiring

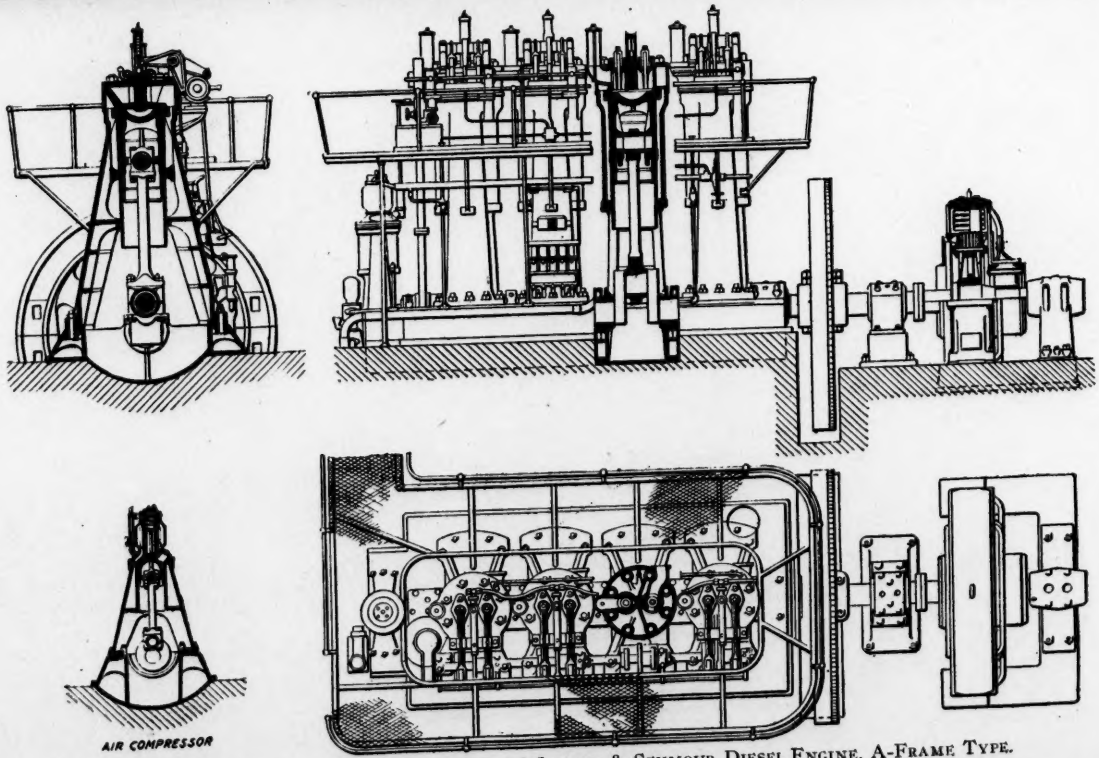


FIG. 1.—PLAN AND ELEVATIONS OF 500-H.P. McINTOSH & SEYMOUR DIESEL ENGINE, A-FRAME TYPE.

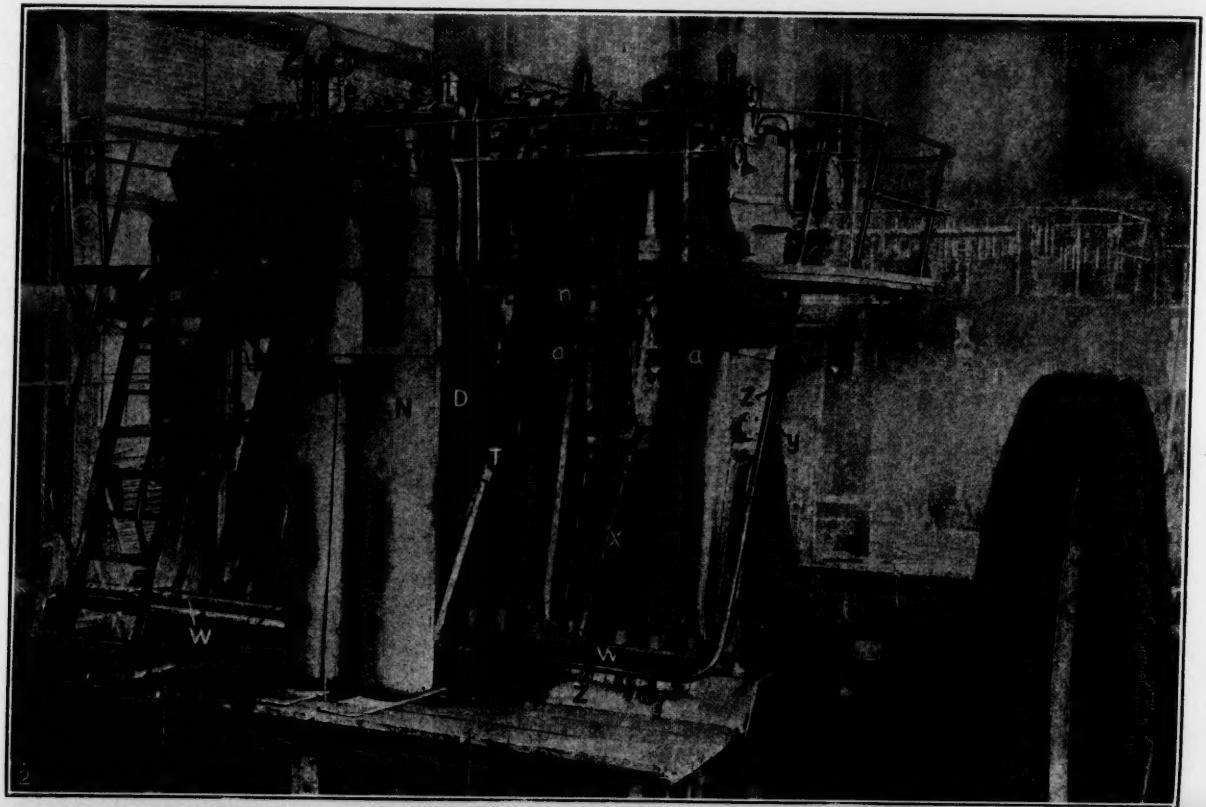


FIG. 2.—FRONT VIEW OF 500-H.P. McINTOSH & SEYMOUR DIESEL-TYPE ENGINE ON TEST BLOCK.

FIG. 2.—FRONT VIEW OF 500-H.P. McINTOSH & SEYMOUR DIESEL-TYPE ENGINE ON TEST BLOCK.

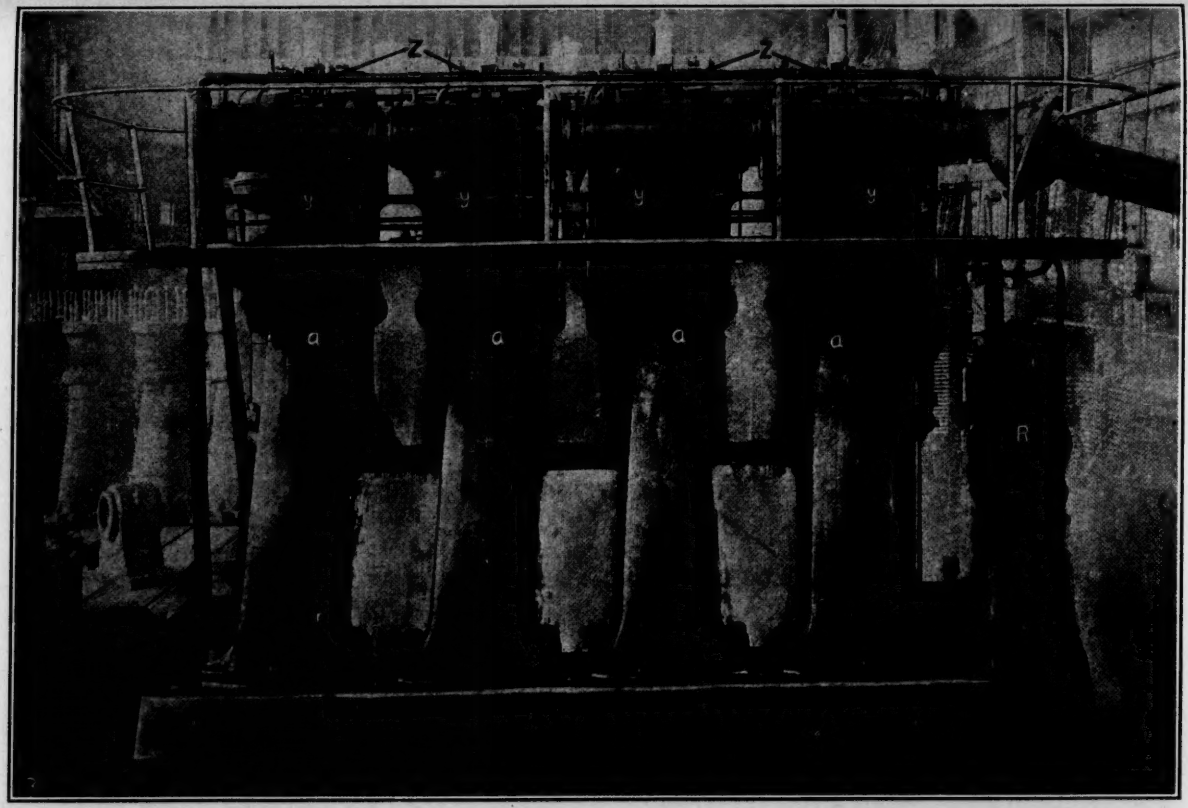
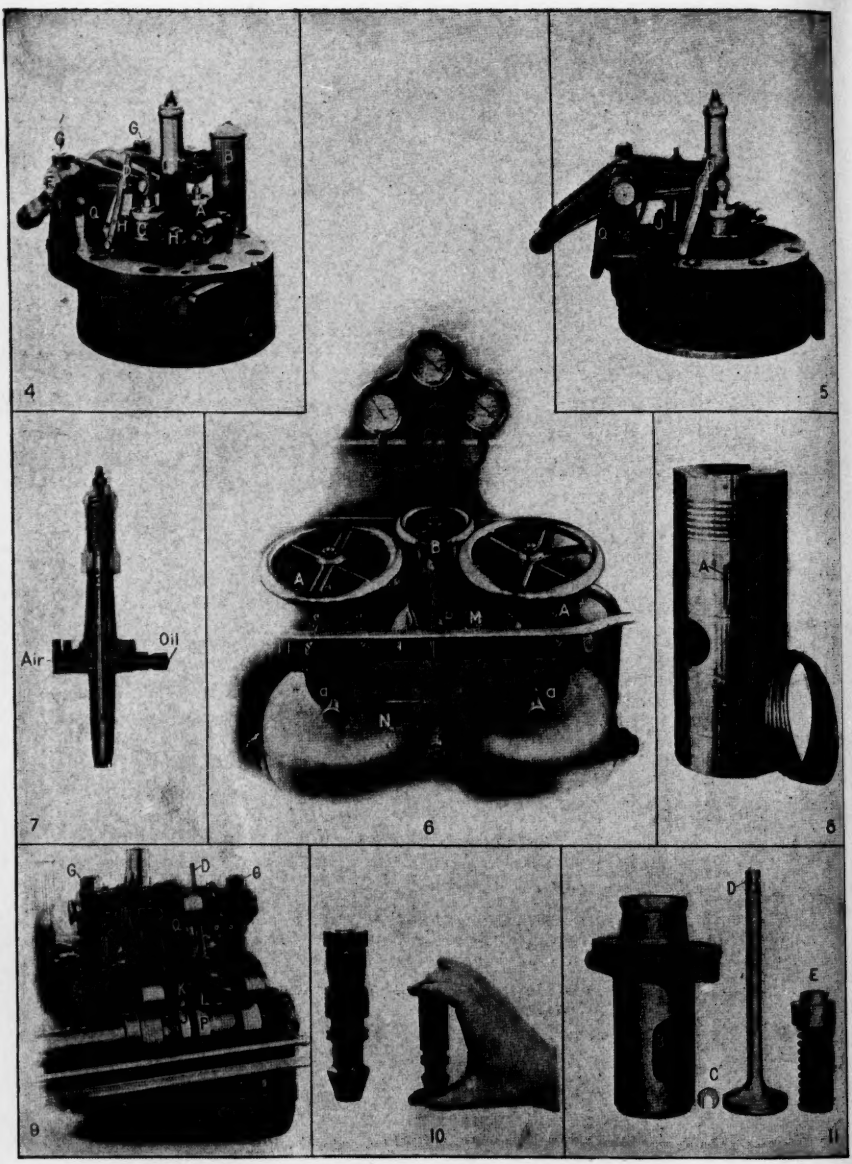


FIG. 3.—REAR VIEW OF 500-H.P. McINTOSH & SEYMOUR DIESEL-TYPE ENGINE ON TEST BLOCK.



FIGS. 4 AND 5.—CYLINDER HEAD AND VALVE MECHANISM. FIG. 6.—AIR BOTTLES AND CONTROL VALVES. FIG. 7.—THE FUEL ATOMIZER. FIG. 8.—THE TRUNK PISTON. FIG. 9.—THE CAM SHAFT AND VALVE OPERATING LEVERS. FIG 10.—DETAILS OF ATOMIZERS. FIG. 11.—THE EXHAUST VALVE AND CAGE.

attention in a steam plant. The comparison becomes much more marked in localities where the steam-plant troubles are increased by bad water.

One of the cylinders (in the engine illustrated the one at the right in Fig. 2 and at the left in Fig. 3) has an additional valve to admit high-pressure air for starting. The top of this valve cage is shown at *J* in Fig. 5, and the cam operating it by the same letter in Fig. 9.

THE PROCESS OF STARTING UP.

When the engine is to be started, all of the exhaust valves are hooked up by the levers *D*, when, compression being avoided, the engine can be easily barred into a position where the piston of the starting cylinder is just commencing its downward working stroke. The exhaust-valve levers are then unhooked. Before doing this the lever *L*, Fig. 9, has been thrown to the right, carrying the roller *K* over to the right-hand side of the cam, where it will be raised by the projection *J* when the crank has passed the center and the cylinder is ready to take air. The valve *A*, Figs. 2 and 6, which controls the flow of air from the starting-air tank *N* to the cylinder through the pipe *n*, Fig. 2, is opened quickly, and air at 750 pounds pressure is admitted to the waiting piston. When the engine starts, the valve *B*, which controls the supply of air to the nozzles through which the oil is sprayed into the cylinders, is opened. As soon as the engine has made about a revolution under the action of this air, the heat generated by compression in the other cylinders is sufficient to ignite the charge of oil that is sprayed in at each working stroke, and when this happens the starting air is shut off by pulling the handle *L*, Fig. 9, to the left so that the roller *K* simply rides upon the concentric left-hand half of the cam and does not raise the valve. The large receiver beside *N* is simply a spare; the compressor delivers into the smaller tank *M*. When the pressure in either of the larger tanks has been lowered by using air from it for starting or by any other cause, the tanks are all put into communication by opening the valves *aab*, Fig. 6, and when they are equalized and the pressure is high enough, both the larger tanks are shut off, tightly bottling up enough air to start the engine many times. The valves on these steel bottles have special discs and seats and are so tight that those sent from Sweden with an engine imported by the McIntosh & Seymour Corporation when considering undertaking its manufacture lost little or no pressure during transit and were used to start the engine when first erected in their shops. The smaller tank *M* remains in action as storage between the compressor and the spray nozzles, its lesser volume being better adapted to variation of the air pressure in accordance with the load.

FUEL INJECTION AND GOVERNING.

For each cylinder there is a separate fuel pump, the four pumps for the engine being assembled in pairs in two right-angled lines. Each pair of pumps is placed end to end, so that the four valves of each pair are fitted in two adjacent chambers in the middle of the pump barrel. In this position they are very accessible and can be removed if the gate that holds down the valve caps has been slacked and swung around. To drive both lines of pumps only a single eccentric is used, the plungers nearest the eccentric being operated directly, while those at the back ends are driven by return rods, a small cylindrical crosshead being fitted both back and front of each line of pumps. Actually, the eccentric that drives the fuel-pump plungers is floating upon another eccentric, and it is by the action of the governor in altering the relative positions of the two eccentrics that the stroke of the pump plungers is lengthened or shortened, thus producing the variations required in the fuel charges fed to the

cylinders. The governor is mounted on the vertical shaft that drives the camshaft.

The fuel pump and governing mechanism represent the only departures in the McIntosh & Seymour engine from the practice of their Swedish associates, who employ the usual European system in governing, which regulates the amount of oil by spilling more or less of the delivery of the fuel pump back to the source of supply by holding the suction valve open for a greater or less period through the action of the governor. The object of this plan is to lighten the work that the governor has to perform and avoid any unfavorable influence upon its action by the reaction of the fuel pump.

THE GOVERNING MECHANISM.

On account of the long and successful operation on McIntosh & Seymour steam engines of shaft governors in which movable eccentrics, the positions of which are directly controlled by centrifugal weights with opposed springs, operate unbalanced gridiron steam valves having a thrust on the eccentric of many times that of the fuel-oil pump of an oil engine, it was decided desirable to adopt a somewhat similar system on the McIntosh & Seymour oil engines of Diesel type. In this the stroke of a separate pump plunger for each cylinder is varied directly by the governor by mechanism that prevents any undesirable reaction on the governor due to the thrust of the pump plungers. This plan requires a governor not materially larger than when the spilling system is used. There is also the advantage that if the governor mechanism becomes deranged with the spilling system it is liable to cause an excess of oil to be fed to the engine, resulting in a dangerous increase of speed, whereas with the McIntosh & Seymour plan the result would be simply to stop the engine.

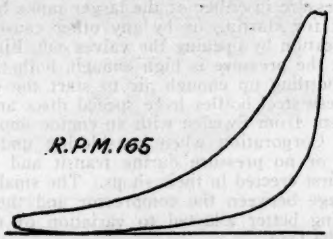


FIG. 12.—DIAGRAM FROM DIESEL ENGINE.

The atomizer used on the McIntosh & Seymour engines is that invented by K. H. E. Hesselmann, of the Aktiebolaget Diesels Motorer. This atomizer has been tried out with various American oils and gives smokeless exhaust and perfect combustion under a wide range of loads, even with Mexican oils as heavy as 14 degrees Baumé. With ordinary oils the excess of atomizer-air pressure above compression pressure runs less than 300 pounds. Its essential feature is that, instead of pulverizing the oil by crowding it down through perforated plates, it draws it by the injector principle into the current of ingoing injection air, which atomizes and absorbs it as fast as it is drawn up. In Fig. 13 the charge of oil is deposited in chamber *C*, which it does not fill, even at an overload. Air is admitted to the chamber *A* and passes through the ports *B*, while it has also access to chamber C through the space *EE*. When the fuel valve opens, the air, rushing through the port *BB*, passes through the expanding passage *F* between the valve stem and the receding wall of the

surrounding fitting and induces by injector action a difference of pressure which causes the oil to flow to the space *J*, into which the oil, having been elevated and broken up through the slotted plates *K* and *L*, is drawn and picked up by the ingoing air. The form of the fuel plate *M* has an important effect upon the efficiency of the atomizer and varies according to the fuel.

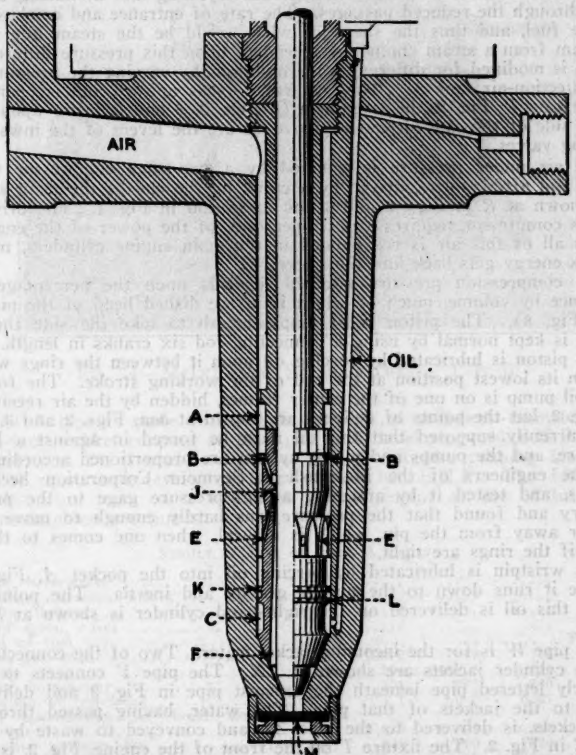


FIG. 13.—THE HESSELMANN FUEL ATOMIZER.

Each of the cylinders in the engine illustrated develops about 125 brake horsepower at rated load and takes about one-third of a cubic inch of oil per working stroke. Such a charge of oil is burned on a 4-cylinder, four-cycle engine once each stroke, or twice each revolution, so that the regulator in apportioning the fuel to the load acts with the same frequency as that of a single-cylinder double-acting steam engine. When it is considered that on tests instantaneous departure from the average of not more than 2½ per cent. and sustained departure of not more than 1½ per cent. from full load to no load are obtained in usual governing and that much closer regulation can be obtained, the sensitiveness of the governor controlling this small amount of fuel is apparent.

This regulated amount of oil is forced into the interstices of the spray

nozzle, Fig. 7, whence it is ejected into the highly heated and compressed air in a finely atomized condition by the air from the tank *M*. This air has a pressure of 700 to 800 pounds, varying according to the density and viscosity of the fuel, also according to the load conditions. As the compression pressure is not over 500 pounds, there remains a pressure difference sufficient to produce a vigorous injecting and atomizing effect even through the reduced passages. The rate of entrance and combustion of the fuel, and thus the shape of what would be the steam line in a diagram from a steam engine, is dependent upon this pressure difference, which is modified for different load conditions by varying the pressure in the injection-air tank *M*. The fuel valve opens upward and is operated by the cam *P*, Fig. 9, the lever arm *Q*, Figs. 4, 5 and 9, being upon the under side of the cam instead of above, as are the levers of the inwardly opening valves.

The air under pressure is supplied by a two-stage compressor with inter- and after-coolers, driven by a crank on the end of the main shaft and shown at *R* in Figs. 2 and 3 and in section in Fig. 1. The driving of this compressor requires about 5 per cent. of the power of the engine, but as all of this air is reëxpanded in the main engine cylinders, much of this energy gets back into the system.

The compression pressure attained depends upon the percentage of clearance by volume, much of which is in the dished head of the piston (see Fig. 8). The piston is of ample length to take the side thrust, which is kept normal by using a connecting-rod six cranks in length.

The piston is lubricated by forcing oil upon it between the rings when it is in its lowest position at the end of the working stroke. The force-feed oil pump is on one of the center frames, hidden by the air receivers in Fig. 2, but the points of delivery are shown at *aaa*, Figs. 2 and 3. It was currently supposed that this oil must be forced in against a high pressure, and the pumps and piping system are proportioned accordingly, but the engineers of the McIntosh & Seymour Corporation became curious, and tested it by attaching a high-pressure gage to the pump delivery and found that the pressure was hardly enough to move the pointer away from the pin, which is natural, when one comes to think of it, if the rings are tight.

The wristpin is lubricated by forcing oil into the pocket *A*, Fig. 8, whence it runs down to the pin by gravity and inertia. The point at which this oil is delivered on the right-hand cylinder is shown at *b* in Fig. 2.

The pipe *W* is for the incoming jacket water. Two of the connections to the cylinder jackets are shown at *XX*. The pipe *Y* connects to the similarly lettered pipe beneath the exhaust pipe in Fig. 3 and delivers water to the jackets of that pipe. The water, having passed through the jackets, is delivered to the pipe *ZZ* and conveyed to waste by the pipe *Z* in Fig. 2. The fixture *T* on the front of the engine, Fig. 2, is the receiver for the open deliveries for the jacket water from the four cylinders, the four exhaust-valve stems and the intermediate and after coolers of the compressor. The attendant can tell by passing his fingers along the opening if each is running at the right temperature.

The frame is made in two forms, the A-type shown in Figs. 2 and 3, and the B-type, which is shown in Fig. 14. Even the joints where the frames go together are scraped and fitted, so that everything is square and in line when bolted up. The pig iron for the cylinders, pistons and other heavy parts that come in contact with the burning oil is imported from Sweden, and the smaller parts, such as valves, which require special machinery and do not involve heavy freight charges, are procured complete from the Swedish builders, of whom the McIntosh & Seymour Corporation is the American licensee.

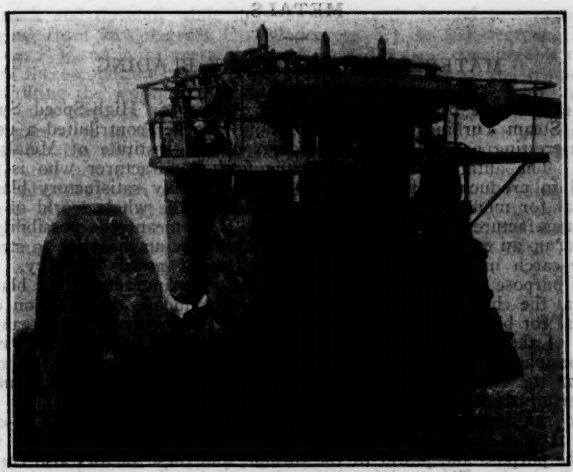


FIG. 14.—MCINTOSH & SEYMOUR DIESEL, TYPE B.

The A type is made with 4 cylinders, all of 18⅞-inch diameter, and 28¾-inch stroke, running at 165 r.p.m. and developing 500 B.H.P. The B-type, Fig. 14, is made with from 1 to 6 cylinders running from 120 to 280 r.p.m. and developing from 40 to 1,000 B.H.P. The following results from tests on a 4-cylinder A-type engine at the McIntosh & Seymour shops are furnished:

RESULT OF TEST ON TYPE A ENGINE.

Rated horsepower					500
Cylinder diameter, inches					18⅞
Stroke, inches					28¾
Rated speed, r.p.m.					164
Kind of fuel					Fuel oil
Load, per cent.	110.5	100.2	77	52	26.2
R.p.m.	162	164	168.5	170	172
Brake horsepower	553	501	385	260	131
Duration of test-hours	0.5	1	0.7	0.67	0.5
Fuel consumption per brake horse-power-hr. in lbs.	0.387	0.393	0.398	0.418	0.565
Injection pressure, lbs.	900	850	800	820	790
Exhaust-gas appearance	Clear	Clear	Clear	Clear	Clear
Inlet temp. of cooling water, deg. F.	70	70	70	70	70
Outlet temp. of cooling water, deg. F.	156	160	160	160	158
Room temp., deg. F.	76	76	75	76	76

Fig. 12 is a reproduction of an actual diagram from the engine. The indicated horsepower at the time of the above tests is not given, but we have seen diagrams which, interpreted under the given conditions of brake horsepower, spring, etc., indicate a mechanical efficiency of over 80 per cent.—"Power."

METALS.

MATERIALS FOR TURBINE BLADING.

Under the title of "Specifications for Alloys for High-Speed Superheated Steam Turbine Blading," Mr. W. B. Parker contributed a useful and interesting paper to the proceedings of the Institute of Metals recently. The author remarked that the first manufacturer who is successful in producing, at a reasonable cost, a really satisfactory blading material for modern impulse turbines will have the whole world of turbine manufacturers as possible customers. This means the possibility of building up an enormous business, and it should be an incentive to strenuous research in this branch of the non-ferrous metals industry. The special purpose of the paper was to place before manufacturers of blading material the chief practical conditions which influence the selection of a material for blading a high-speed turbine driven by superheated steam.

One of the most interesting portions of the paper is that which deals with peripheral speeds and stresses in the blading. The author points out that the rotating portion of a reaction turbine is usually a drum or cylinder of steel, and is termed the drum or rotor. The rotating portions of an impulse turbine are usually wheels or discs of steel—commonly termed bucket wheels. The speeds at which these revolve vary with the purpose and power of the turbine, and these factors also control their diameter. The peripheral speeds are therefore very variable in different machines, and depend upon the ratios of the above factors. The peripheral speeds are the direct cause of the stresses to which the blading is subjected when actually working, and are consequently the primary reason for the demand for strong material for blading. These stresses are borne by or located in that portion of the blade by which it is fixed in the wheel or drum—they are not evenly distributed over the whole section of the blading as it is manufactured by the blading manufacturer. This is not the occasion to go into the question of the various methods used for fixing the blades into the rotating parts, but it is necessary for a correct realization of the specific requirements of turbine manufacturers in this matter to give at least one typical example of every-day practice, which is termed the "dovetail" method. In this the shoulders of the dovetail are under compression and shear stress, and the neck is under a tensile stress.

The tension stresses in the neck of the blades may vary from, say, 4,480 pounds (2 tons) per square inch to 15,680 pounds (7 tons) per square inch of the area of the neck. It depends somewhat upon the conditions of the contract covering the design. It might be noted that some contracts specify 25 per cent. overspeed tests, which increase the stresses on the blades about 50 per cent. It is important to notice that the neck is formed within the thickest portion of the blade cross section, and therefore, comparatively speaking, of the weakest portion of the section. This point is common to many other systems of fixing blades. Consequently, when turbo-blading specifications specify strengths, &c., they intend that the thickest portion of the section shall possess the strength and other properties specified. It is no use suppliers complaining that this is unfair, difficult, and so on, for it is an absolute necessity, and a practical condition which must be met. The most suitable shape and dimensions for the dovetail of each size and design of turbine has been decided by a long series of tests, using actual sections made up with all the various blading materials so far offered by manufacturers. There is no need to detail these, as they are of no immediate importance to blading manufacturers.

It is of the utmost importance to bear in mind that during the whole working life of the turbine the stresses are imposed upon the material of

the neck and dovetail while it is simultaneously heated to temperatures ranging from 26.6 degrees C. (80 degrees F.) to 399 degrees C. (750 degrees F.), and it is imperative that it should possess considerable rigidity under these conditions. The exact combination of stress and temperature constituting the working condition varies not only from machine to machine, but also, as previously mentioned, with the position the blades occupy relative to the steam inlet and exhaust outlet. The general condition is, that the longest blades are the highest stressed, and the most moderately heated, because they are only required for the low-temperature (and pressure) end of turbines. The shortest blades are the highest heated and the most moderately stressed, but progress in turbine manufacture is now demanding that they shall carry still higher stresses at these higher temperatures.—"Shipbuilding and Shipping Record."

CASE-HARDENING PROCESS FOR TOOL STEEL.

HIGH TEMPERATURES AND SHORT CUTS TO BE AVOIDED—INFLUENCE OF TIME AND TEMPERATURE UPON RESULTS—CASE AND CORE TREATMENT.

By R. A. Millholland.*

The merchant mechanic is constantly using tools made from case-hardened steel and an intelligent comprehension of some of the problems involved in the hardening process will assist him in the choice of his equipment.

There is an axiom that always comes into my mind whenever the subject of case hardening is mentioned, and it is this: The depth of carbon penetration in case hardening depends on the time of heating and the temperature at which the work is heated. This is not a new theory in case-hardening practice by any means, for it has been known for a considerable number of years by a very few clever men who did not see fit to talk much about the matter, preferring to keep it as a trade secret. Another good axiom to heed is, high temperatures produce poor results. By high temperatures I mean temperatures above 1,700 degrees F. The proper case-hardening range lies between 1,600 and 1,700 degrees.

There are an infinite number of reasons why the higher temperatures should be avoided in case hardening. The greatest reasons are: First, the use of temperatures above 1,700 degrees F. for case hardening will cause too much carbon to be absorbed in the carbonized area and a supersaturated case is the result; second, the high temperatures will cause excessive crystallization of the steel that is being carbonized and serious difficulty will be experienced in refining both the case and the core. Between 1,600 and 1,700 degrees F. the core structure is not very materially changed by the ordinary heating time, which on a rough average is about 4 hours. Crystallization does not take place to a certain extent, but the refined structure is quickly restored by careful treatment. The lower carbonizing temperatures minimize the possibility of the pieces warping during carbonizing. Temperatures below 1,600 degrees F. are, as a general rule, impractical from the economical standpoint, if for no other reason. The speed of penetration is so much greater at 1,600 to 1,700 degrees F. that it is impractical to use the lower temperatures where cost is an item to be considered. The depth of penetration varies in a direct ratio with the temperature; that is to say, at 1,700 degrees more carbon penetration will be secured in a given time than will be secured at 1,600 degrees F. in the same length of time.

*Consulting metallurgist Millholland Tool Company, Indianapolis, Ind.

The time and temperature factors are very important in case hardening, and to strike a happy medium of temperature and time is at once both highly desirable and very difficult. A disregard of either factor will result in disaster. On several occasions I have been asked why certain steels seemed to burn out when they had been subjected to long carbonizing heats. Never in all my experience did I find the steel to be the real cause, in almost every case it was due to the use of temperatures above 1,700 degrees F. Generally coupled with the overheating, the practice of dumping the parts directly from the pots was employed. Both are very poor practice, but the combination of the two is bound to give some mighty poor results. A certain manufacturer was trying to defend the practice of dumping directly from the pots, but he admitted that his product gave 100 per cent. more service when reheated to refine the core, and he showed me several test records to that effect.

The writer could cite innumerable incidents to prove the superiority of complete heat treatment over the short cuts that are so often used to economize on case-hardening expense. "Safety First" signs are conspicuously displayed in almost every shop in the country, and yet the manufacturer, in some cases, still clings to the before-Noah-and-the-Ark practice of quenching the carbonized pieces directly from the pot, and calling the case-hardening operation complete. It is always best, when at all possible, to allow the parts to cool in the pots before they are removed from the carbonizing compound. There is a dual reason for doing this. The first is that the carbonizer is saved from oxidation by the air, which would render it almost useless for further service, and the second reason is that on heats longer than 4 hours the crystalline structure of the core is so coarsened that the hardened part will not have the strength consistent with the factor of safety, upon which principle all conscientious manufacturers build their products. However, on heats of less than 4 hours, it is generally the case that the parts can be quenched from the pot and then treated to refine the case, which will give a result that will fill all but the most exacting requirements. The one objection to this practice as applied to this particular method is that the carbonizer can only be used once before it loses much strength. The reason is that the air, containing as it does a large per cent. of oxygen, oxidizes the carbon in the carbonizer and makes the carbonizer useless for further work. If, however, a cheap carbonizer is used, similar to one of those I have already mentioned, a material saving in actual carbonizing cost can be effected.

On work of a larger nature which requires a longer heating time than 4 hours it is essential to treat both the core and the case. Especially is this true of parts that are subjected to shock and vibration. To the uninitiated it is astounding to see what the actual difference is between a properly and an improperly treated case-hardened part. It is safe to put the difference in strength at a minimum of 50 per cent. for both the tensile strength and the resistance to shock.

Now let us consider the most practical method for treating both the case and the core to secure the best results. After the piece has been carbonized the first object is to refine the core. The practice in all cases is the same, whether carbon or alloy steels are being treated. The only variation to be considered is the critical temperature of the original steel as received from the mill prior to the carbonizing operation. In the standard mill specification open-hearth, case-hardening stock the critical temperature is very close to 1,650 degrees F., and is the one at which the core of the case-hardened part should be treated.

There are many different methods for doing this, and I shall only touch briefly on the two most important ones. For rough work that is to be ground all over after hardening, or for work that will allow a slight amount of distortion and scale, I believe heating in the open furnace to

be the most economical method that can be employed. A large amount of work can be handled rapidly, materially reducing the amount of labor per piece. The ordinary commercial heat-treating furnace can be used to the best advantage. By a pyrometer or some other device the temperature of the furnace should be regulated to 1,650 degrees F. and the furnace charged with the parts to be treated. It is essential that the temperature of the furnace be regulated as nearly correctly as possible in order that the steel is not overheated. When the parts are thoroughly saturated with heat and the pyrometer indicates that the heat is what it should be, the work should be quenched in a heavy oil. From careful experience and experimenting I find for quenching from the higher heats employed in heat treatment that an oil having a specific gravity of 22 Baumé, a flash point of 420 degrees F. and an ignition temperature of 480 degrees F. is suitable. These specifications conform to practically all fire-insurance regulations, and at the same time the oil can be obtained at considerably less than the many special oils sell for.

TREATING CORE.

The other method for treating the core is used chiefly on transmission gears and ordnance equipment and anywhere that distortion and scaling must be held to a minimum. The lead bath is used in this method to heat parts. The lead is heated in crucible furnaces designed for that purpose and the parts are immersed in the molten lead and held there until they are thoroughly heated and then quenched in practically the same manner that the furnace-treated pieces are. The advantage of this method is that the pieces are uniformly heated in a bath that excludes all air and prevents oxidation. The result is that the parts come from the oil free from scale and distortion is reduced to a minimum. In both methods the work should be transferred from the furnace to the oil-quenching tank as quickly as possible to secure the greatest strength in the core.

CORE CASE.

Following the treatment of the core the case should receive even more careful attention than the core. A hard surface is the primary object of case hardening, and unless the case has the desired hardness, no matter how much care has been bestowed on the core the effort is wasted if the case is not hard enough to fill requirements. The essential principles of treating the case are identical with those involved in treating the core. The work is heated to the critical temperature of the case, which in open-hearth steel will be about 1,400 to 1,425 degrees F., and this time it is quenched in fresh water, or where extreme hardness is desired it is quenched in brine. The lead pot is used extensively where the work must be free from scale, but work that is to be finish-ground all over can be very satisfactorily heated in the ordinary furnaces. It is always a good policy to run test pieces through with each lot so that there is some means of checking the results that are being obtained. The test piece should show at least 5 degrees deflection before a complete fracture is obtained. The fracture should show a soft core of a moleskin-like texture, and the case should look like the fracture of a file or that of nicely hardened tool steel.

ALLOY STEELS.

The same principles of heat treatment apply to alloy steels as they do for plain carbon steels. The only differences are that the critical temperatures of alloy steels vary somewhat from those of carbon steels, and it is the general rule that alloy steels will harden quite satisfactorily in a light quenching oil. This last feature has a strong tendency to minimize

cracking during hardening. In every instance the up-to-date steelmakers can better determine what are the best temperatures at which to treat their respective brands, but no matter what the make of steel is, be it carbon or alloy, the principles governing the case hardening of it are the same the world over.—"Metal Worker, Plumber and Steam Fitter."

MATERIALS USED IN CASE-HARDENING.

By R. A. Millholland.

Of all the forms of heat-treatment no doubt case-hardening is the most extensively employed. It is merely an application of the old cementation process of making steel from iron. Of course, improvements have been made over the original methods, both in the process and in the materials involved. There are two prime requisites that case-hardening fulfils more efficiently than any other process known to modern mechanical science—namely, a very hard non-wearing surface and tough malleable core that is capable of absorbing an enormous amount of vibration without serious difficulty.

The application of the process of case-hardening is too expensive for exhaustive enumeration in this limited space. Generally speaking, short shafts, pins, bolts, collars, thrust rings, screws, bearings, spindles, gears, ratchets, clutch dogs, or any machine parts that are made of steel and are subject to wear can be advantageously case-hardened at a very small cost. The writer has in mind a company that makes a practice of case-hardening all the steel parts of its tools and jigs used in the manufacture of automobiles. The maintenance on these tools is surprisingly low. Locating lugs and centering blocks do not wear away on the edges and become inaccurate. Practically the only replacement that is done on the tools at all is of drill bushings and boring-bar guides. The locking devices, set screws, and clamps showed no sign of wear after months of constant production. One small cotter-pin hole has had 40,000 shackle bolts pass through it, and it is still as accurate as micrometers can measure. The same jig was replaced three times in drilling 20,000 holes before the case-hardened jig was made. No change in design was made; only the case-hardened feature was added. Any steel surface, whether a bearing one or not, that is liable to be nicked or marred by rough handling can be made practically indestructible by case-hardening. The advantage of the proces of case-hardening low-carbon steel in place of using high-carbon tool steel is obvious in many instances, yet it must be clearly understood that case-hardening is not a cure-all skin plaster to be used indiscriminately for any and all purposes. Each firm must learn largely from experience what can be case-hardened to advantage.

Let us first have clearly in mind what is meant by the terms case-hardening and the cementation process. We must hark back to the days when the art of making steel was new; how far back no one really knows, but the names of Damascus and Toledo are familiar to all readers of history. Were you to ask one of these readers what was the most striking thing that he remembered about these two ancient cities, no doubt the answer would be: "The swords of Damascus and blades of Toledo." Without a doubt both of these famous steels were made by the cementation process with some improvements, the secrets of which died with the prestige of the cities.

A fine grade of wrought iron was the base upon which the cementation process was founded. Flat bars of iron were placed in furnaces packed in layers of charcoal and subjected to a temperature in the range of 1,400 degrees to 1,650 degrees F. The carbon in the charcoal was dis-

solved by the iron, and when carbon had penetrated the steel and thoroughly saturated it, the cementation was considered complete and the bars were removed and "stack-welded" one upon the other to a homogeneity that has been the envy of succeeding generations of steel craftsmen. The ancients realized the value of "working" their steels, and the results secured were remarkable. The writer does not want to be misunderstood, however, causing some to believe that ancients could produce better steel than our latest modern manufacturers do. The ancients had nothing to surpass or even equal the chrome-vanadium steel of today.

The process of case-hardening, then, is nothing but the proper regulation of the cementation or infusion process, as some see fit to call it. The old method of using charcoal as a means of injecting, or infusing, the carbon into the iron is slow and costly. Many newer and better materials have been discovered which are far superior to the old charcoal process.

It was discovered some years ago that the presence of nitrogen in the carbonizing material greatly increased the speed of penetration or infusion of carbon into the surface of the iron. It was found after investigation that nearly all of the numerous patented carbonizing compounds have only two really essential elements in them, and despite the claims of the virtues of their respective products, carbon and nitrogen still remain the two major elements involved in case-hardening. It is of little or no consequence to the practical man whether the carbon penetrates the steel in a gaseous or in a solid state. Let the ultra-scientific theorize on that subject, and meanwhile let us consider the natural carbonizers and their faults and virtues. Among the natural carbonizers, those most extensively used are granulated bone, charred hoofs and leather, beet-sugar pulp, and crude raw sugar. Granulated bone is no doubt the most widely used of all carbonizers, and while it has some almost unforgivable evils, its availableness and price make it a universally used carbonizer. Its one and only great drawback is the high phosphorus content, which has a noticeable effect on the toughness of the case-hardened area, making it brittle and prone to chip or flake off in service. However, after bone has once been burnt over and still contains a high percentage of carbon, the danger from the phosphorus decreases with the number of times the bone is used. I should say that 36 hours of actual carbonizing heat is all that bone will stand and give any kind of results at all. If the bone is properly fortified after each heat, it can be used repeatedly, and grows better in carbonizing quality after each fortification. Highly uniform results can be obtained by uniform methods in fortifying the "spent" bone. By fortifying I mean restoring the carbonizing power of the bone. Bone is made up largely of lime, which really forms the body or structure of the bone, and is in evidence when, after repeated heatings, small bits of chalky white substance are found in the mass of burnt-over bone, indicating that the bone is almost spent or burnt out. The writer has developed a method of his own regarding this process of refortifying spent bone, and will gladly furnish any fellow-craftsman with more particular details than this limited space will permit.

Charred hoofs and leather, when mixed with hickory charcoal, make a very good carbonizer, but the mixture is very uncertain, and where uniform results are required the fortified bone is best, and costs much less. Beet-sugar pulp is obtainable from any sugar refinery at a surprisingly low cost, and in the writer's estimation has no rival for case-hardening nuts and screws, as the work comes from the carbonizing pots without any adhering dust particles and fused bits of metal that we often find in the charred leather. Five hours is the longest heat that beet-sugar pulp will stand before it is spent, and nothing is left but a light, flaky dust in the cans. The shrinkage of the pulp is at least 30 per cent. Beet-sugar pulp is high in volatile carbonizing gases, and for that reason

the carbonizing retorts must be thoroughly gastight to secure the depth of case in the parts undergoing case-hardening.

Now let us consider the artificial, or rather the manufactured, case-hardening compounds. Some are patented, but many are not, and I shall describe several different brands. Each manufacturer of carbonizing compound quite naturally claims his to be the best, and I have pamphlets and booklets on my desk that show clearly and quite convincingly, I confess, that the particular brand of carbonizer exploited in the pamphlet is the best obtainable. I might truthfully say that I have not run across any real bold frauds in this particular line,—at least, not within the last six years,—and any of the better-known products are safe and reliable carbonizers to use—that is, disregarding the economical side of the question. I have found all of the present carbonizing compounds put out by the manufacturers to be very expensive, and they all lose their body and disintegrate into fine dust, and do not respond to fortifying as well as the natural bone does. For very fine work I strongly recommend the use of a manufactured carbonizer, because the results obtained are undoubtedly the superior of bone carbonizer, but for ordinary purposes bone is quite satisfactory.

The reason for this discrimination between ordinary work and delicate case-hardening is based on scientific principles. The writer has found through his own experience that a carbon content in the case of 0.85 per cent. gives the very best results for a hard, stiff case that will neither wear away easily nor chip off under a slight blow. Of course, it is quite impossible to secure such a distinctly theoretical carbon content, but anywhere from 0.85 to 1 per cent. carbon in the case will give remarkable results if properly heat treated. A closer adherence to this range is secured by using the balanced compounds of the carbonizer manufacturers rather than by indiscriminately using natural carbonizers. However, for the most part, the fortified bone will give very good results, and it is much more economical.—"Iron Age."

ELECTROLYTIC COPPER.

Copper is a material in which the electrical engineer is especially interested. We do not know what proportion he uses of the million tons a year which is produced in the world in normal times, but it is certainly a very large one. In many ways his specifications for copper have to be more stringent than those of anyone else, and as a consequence he has a peculiarly direct interest in the development of electrolytic refining. It is this development which has given him a commercial copper of some 2 to 3 per cent. better conductivity than Mathiessen's standard. According to Mr. Burkewood Welbourn, some 70 per cent. of the copper production of the world is now refined electrolytically; and, as a specific figure, 461,583 tons were so refined in the United States in 1907. The importance of this development, from the electrical point of view, was well illustrated by Professor Marchant in his recent address to the Liverpool Engineering Society, in which he pointed out that traces of arsenic in copper reduced the electrical conductivity by 40 per cent., and 1 per cent. of phosphorus reduced it by 30 per cent. What is to the electrical engineer an incidental advantage of electrolytic reduction is that the gold and silver which are associated with copper ores can be recovered instead of being lost, as they were with earlier methods. Referring to the figure already given for the production of electrolytic copper in the United States in 1907, it may be mentioned that 13,995,436 ounces of silver and 272,150 ounces of gold were recovered in connection therewith.

In extracting copper from the ore it is usually necessary first to treat the ore in calciners to get rid of part of the sulphur before smelting. After this it is smelted to extract the earthy matter and to concentrate the copper and other valuable metals with part of the sulphur and iron into a matte. This matte will usually contain from 45 to 50 per cent. of copper. It is transferred to a Bessemer converter, which blows it up to a content of from 97 to 99 per cent. of copper with any silver or gold which may be present. A material of this composition may be obtained by these processes from an ore containing only 3 to 4 per cent. of copper. The copper is run from the converter into slabs, which are then transferred to a reverberatory furnace, with a capacity of 150 to 200 tons, for further purification. The copper is then run into molds, to form the cast anodes, ready for the electrolytic vats. The anodes are usually 36 inches by 36 inches by 1½ inches thick and weigh 500 pounds each. According to Mr. Burkewood Welbourn, the cost of converting a 50 per cent. matte to metallic copper, under good conditions in the United States, is about ⅓d. per pound of refined copper. This figure is taken from the chairman's address, which Mr. Welbourn delivered to the Manchester Section of the Institution of Electrical Engineers on the 16th of November, 1915. The essentials of the brief description of copper refining which we give here are from the same source.

In the electrical part of the process, Mr. Welbourn stated, most firms employ the multiple system, in which anodes of cast copper and cathodes of electrically-deposited copper, 36 inches by 36 inches by 1/32 inch thick, are hung in lead-lined vats. The space between the electrodes is 1½ inches to 2 inches, and the pressure employed is about 0.3 volt for every pair of electrodes in series. The electrolyte is an acidulated solution of cupric sulphate, and contains 1½ pounds to 2 pounds of $CuSo_4$ and from 5 ounces to 10 ounces of H_2So_4 per gallon of water. It is maintained at a temperature of 130 degrees F. The current density is usually 7½ to 12 ampères and should not exceed 18 ampères per square foot of anode surface, as it has been found by experience that the mechanical properties of the metal depend on the current density. With the 18-ampère figure a high temperature and a rapid circulation of the electrolyte must be maintained. In practice a 1½-inch thick anode decreases to ½ inch thick in ten days, and pure electrolytic copper is deposited on the cathodes until they are just over ½ inch thick. The precious metals settle as slime at the bottom of the vat. The ½-inch thickness of anode which is left is removed to anode furnaces for re-melting and further casting. Theoretically, with 100 per cent. current efficiency and 0.3 volt between electrodes, 1 pound of copper should require about 0.115 kw. In practice the gross kilowatt-hours required for refining and for power and light in the largest works is from 0.14 to 0.15 per pound. The over-all cost of electrolytic refining is not published by the refiners, but Mr. Welbourn believes it to vary from ¼d. to ⅜d. per pound.

The finished cathodes from the vats are melted down, blown with air, poled, and then cast into wire bars, the average weight of which is 140 pounds in England and 220 pounds in the United States. If required by wire manufacturers, bars up to 800 pounds in weight can be produced. For electrical purposes the copper is mostly required in the form of rolled strip or wire. For wire-drawing, the bars are heated to a bright red, broken down in rolls, and afterwards rolled into rods in a rod mill, and coiled while still hot. During these processes the copper oxidizes rapidly, and it is necessary to remove the scale by pickling. The copper in the scale is afterwards recovered by an electrolytic process. The cleaned coils are then taken to wire-drawing machines, and drawn down through chilled iron or diamond dies, the latter being used for the smaller sizes. The hardness or softness of the wire produced depends on the

number of dies through which it is drawn. Speaking generally, the greater the number of dies the harder and stronger the wire. Really soft wire cannot, however, be directly made in this way, and for its production the drawn wire has to be annealed. The modern way of carrying out this work, known as the Bates and Peard process, consists in the use of a cast-iron retort, heated externally, and with two descending mouthpieces which dip into a water-seal. The retort is filled with steam at a temperature of 800 degrees F. for small wires and 1,100 degrees F. for larger ones, and the wire to be annealed is slowly fed through the heated retort. For producing long lengths of wire it is frequently necessary to join wires together during manufacture. For small sizes, used in the construction of insulated cables, Mr. Welbourn states that the work may conveniently and economically be carried out by the use of electric butt welders, and that joints formed in this way will stand further drawing through dies. For trolley wires, some engineers prefer that wire bars of such a weight should be used that the lengths required are produced without any joints. An alternative method is to joint the rolled rods, from which the wire is to be drawn, by scarfed joints, which are brazed together with silver. Joints formed in this way are entirely successful, and will stand drawing down. In the finished wire the joint cannot usually be detected by the eye, and its tensile strength is not less than that of the rest of the wire.

A large part of Mr. Welbourn's address was concerned with the properties of copper conductors. He pointed out that the present British standard for such conductors, issued by the Engineering Standards Committee in March, 1910, was understood to be under revision, and that a new standard had been introduced by the International Electro-Technical Commission, publication No. 28, of March, 1914. This latter standard had been adopted by the American Institute of Electrical Engineers, and it was to be hoped, in the interests of uniformity, that it would also be adopted by the Engineering Standards Committee. Mr. Welbourn would, however, like the standard definition to be extended by a clause stating that the resistance of hard-drawn copper should be considered 2.7 per cent. greater than that of annealed copper. The present Standards Committee's definition of hard-drawn copper wire is wire which does not elongate more than 4 per cent. on a length of 10 inches when broken by tension, and which has a resistance 2 per cent. higher than annealed copper at 60 degrees F. The Board of Trade's regulations for overhead lines require the elongation to be not less than 2 per cent. on a length of 10 inches. In this connection Mr. Welbourn points out to us that no satisfactory definition of hard-drawn copper exists, and that no definition can be satisfactory which does not cover tensile strength, elongation, limit of proportionality, modulus of elasticity (in exceptional cases), and ohmic resistance.

Dealing with the testing of hard-drawn copper, Mr. Welbourn stated that in practice it was found that the actual tensile strength increased proportionately to the diameter, and not to the sectional area, although the tensile strength in tons per square inch increased with a decrease in diameter. As a result of the analysis of a number of tests, Mr. A. P. Trotter found that the tensile strength could be expressed by a formula of the form $T = a - b\ D$, a and b being constants. This formula by itself, of course, gives little information about the wire, as it does not cover the extension. For minimum extension Mr. Welbourn stated that tests or wires from 0.075 inch to 0.5 inch in diameter gave the expression $E = 5\ D$. Working with these results, Mr. Pye, in the Journal of the Institute of Metals, had suggested the following as a definition of hard-drawn copper: "Hard-drawn copper when in the form of circular wires shall have a tensile strength not less than that given by the formula

$T = 30 - 20\ D$, and an extension per cent. on a marked 10-inch length, including point of fracture, of not less than that given by the formula $E = 5\ D$." An examination of the standard figures prepared by the Wire Committee of the American Society for Testing Materials (1909) for wires ranging from 0.04 inch to 0.46 inch diameter showed that they agreed closely with Mr. Pye's specifications for strength, and that the minimum extension figures agreed with the formula $E = 4\sqrt{D}$. Mr. Welbourn thought Mr. Pye's definition might be adopted by the Engineering Standards Committee. It called for a high standard, which could easily be obtained by the leading manufacturers throughout the world.

In view of the extensive use of copper overhead conductors on tramways, &c., and the importance of the elastic properties of the wire in relieving strains, it seemed desirable that any definition of hard-drawn wire should include a reference to the limit of proportionality (the elastic limit), and at first sight to its modulus of elasticity. There were difficulties, however, in making these delicate measurements in works routine. The limit of proportionality varied from 50 per cent. for wires of 0.08 inch diameter to 70 per cent. for wires of 0.05 inch diameter when measured on 10-inch samples. The Engineering Standards Committee might stipulate for a minimum limit of proportionality for all hard-drawn wires, and state the conditions under which the test should be carried out. An excellent basis for the matter was to be found in the Standards Committee's Report No. 55, which dealt with an investigation carried out by the National Physical Laboratory on some standard sizes of wire used by the Post Office. Mr. Welbourn suggested that the Research Committee of the Institution of Electrical Engineers would confer a great boon on the industry if they would carry through an investigation on similar lines for all standard circular wires from No. 5 S.W.G. to 7/0 S.W.G. The values could then be interpolated for all other sizes of wire by using a large-scale chart which embodied the results of the two investigations. Mr. Welbourn would also like the investigation to be extended to stranded conductors. From the small amount of work he had done in the matter there seemed reason to believe that the limit of proportionality was higher for a given section of stranded conductor than for a corresponding solid conductor. On all strands containing more than three wires a considerable number of tests showed that the effective strength should be taken as 10 per cent. below that of a similar number of straight wires. This appeared to be due to the difficulty of getting the strain evenly distributed on the wires of a strand by any form of grip, and to the fact that the layers of wires were of unequal length.

On the question of the modulus of elasticity Mr. Welbourn remarked that an examination of the stress-strain curve of any hard-drawn copper wire showed that the modulus must be constant for any particular size, since the curve was a straight line up to the elastic limit, but the modulus was not necessarily the same for different sizes of wire. In making measurements for true modulus values it was, in his opinion, necessary to work on long lengths of wire, which approximated to the spans employed in a tramway or other overhead line. He had been led to that conclusion by a study of the results recorded by the National Physical Laboratory, in the Engineering Standards Committee's report, No. 55, which dealt with lengths up to 50 feet. Mr. W. B. Woodhouse had made similar tests in spans of 110 feet. The moduli given in the report varied between 17,200,000 and 18,200,000 on the second and third applications of the load, and particular attention was drawn to the different values of the readings on the first and subsequent applications of the test load. On the first application the kinks in a coil of wire were pulled straight, and the apparent moduli varied from 12,300,000 to 15,300,000, while on subsequent loading the values were, as stated, 17,200,000 to 18,200,000. In the report

it was said: "It is thought, however, that the value of the apparent moduli will be of interest and use, since it is these which must be taken into account in the erection of long lengths of wire." Mr. Welbourn wished to express strong disagreement from this statement, because on all well-constructed overhead lines the wires should first be strained to a tension higher than that at which they would be permanently bound in. The object was to get rid of the kinks referred to in the report. The procedure made it permissible to take advantage of the real modulus value. He thought, in view of the evidence, that it was permissible to adopt an all-round modulus value of 18,000,000 for future work instead of the usual text-book 16,000,000, but thought it not practicable to embody a minimum modulus value in ordinary specifications of hard-drawn copper.

The only other point in Mr. Welbourn's address we can mention had reference to the tensile strength of hard-drawn copper wire. In connection with the widely-held belief that the strength of such wire lay in the skin being hardened during the drawing process, Mr. Welbourn investigated the matter by taking a length of 0.4 inch diameter hard-drawn circular trolley wire and running six pieces down to different diameters with such care as to avoid undue heating of the specimens. The pieces were then tested, and it was found that there was but little difference in the breaking stress in tons per square inch. It varied from 24.7 to 25.8, and, on the whole, gradually rose as the pieces became smaller. He pointed out that the result was not absolutely conclusive, since the work done in the lathe would have some effect in hardening the skin at each stage. Another experiment to determine if the strength of copper wire was uniform throughout its cross-section was made by taking a piece of 0.25 square inch grooved trolley wire and reproducing the wear on the face, due to the trolley, by gentle filing. Tests were made at various cross-sections and showed but little variation in the maximum stress per square inch for the different conditions. Mr. Welbourn also carried out tests on the effect of temperature on hard-drawn copper wire using 4/0 trolley wire. His results confirmed those of the National Physical Laboratory which show that "the reduction in breaking load due to temperature appears to be approximately one-tenth of 1 per cent. of the breaking load at normal temperature per degree Centigrade rise of temperature."—"Engineering."

PROCESSES OF ELECTRIC WELDING.*

By C. B. Auel.†

SYNOPSIS.—Describes various processes of electric welding. Boiler-tube welding a good field for the application of the Slavianoff process, as shown by the Pennsylvania R. R. Comparison of costs of smiths' welding and electric welding on similar pieces.

[The growing application of welding to the repair of power-plant equipment makes it important that the engineer become familiar with the different processes.—Editor.]

Electric welding can be accomplished by any one of several processes, but none is of universal applicability; that is to say, each one is more or

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less limited to a certain field as well as to a given range in such field. Furthermore, there is little advantageous overlapping of their respective fields, so that the processes are supplementary to one another rather than complementary.

There are three clearly defined processes—arc, incandescent and electro-percussive—of which the last mentioned is recent. The first two are capable of subdivision—the arc into the Zerener, Benardos and Slavianoff processes; the incandescent into the La Grange-Hoho and Thomson processes. The differences between them may be stated as follows:

Arc (Direct Current).

Zerener: Arc drawn between two carbon rods which form the electrodes and directed on metal to be welded.

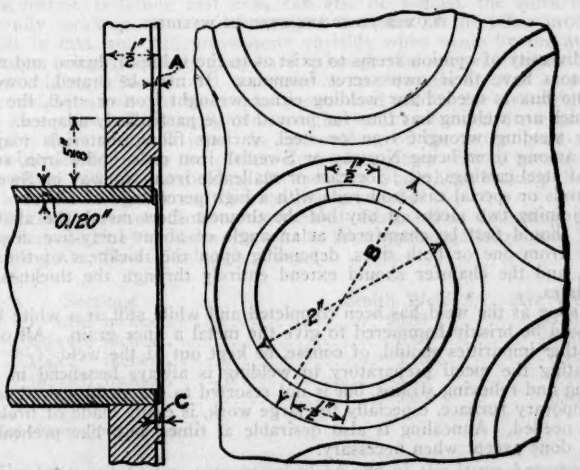
Benardos: Arc drawn between carbon rod, which forms one electrode, and metal to be welded, which forms the other electrode.

Slavianoff: Arc drawn between metal rod, which forms one electrode, and metal to be welded, which forms the other electrode.

Incandescent (Direct Current).

La Grange-Hoho: Metals to be welded form one electrode of acidulated bath; large conducting plate forms the other electrode. Metals butted together after reaching incandescence.

Thomson (alternating current): Metals to be welded form the electrodes; butted together under pressure, causing high resistance contact with heating.



A- *This cut only made deep enough to clean sheet of all scale and make right angle at edge of flue hole*
B- *From 2 1/2 to 2 3/4 in. depending on condition of sheet.*
C- *From 0 to 1/16 in.*

How Tube Plate is Prepared for Welding Tubes.

Electro-Percussive (Direct Current).

Metals to be welded form the electrodes; brought suddenly into percussive contact and vaporized.

The arc processes are autogenous in that welding can be accomplished without pressure, simply by allowing the metals to melt under the influence of the electric current, then to mix and unite as they cool; the incandescent and electro-percussive processes, however, invariably require pressure as a necessary adjunct to their successful accomplishment.

The carbons should be solid, not cored; should neither crack, crumble nor spindle, and should preferably be "graphitic" rather than "common," as the former type is of higher electrical as well as thermal conductivity.

Metal electrodes will range from ⅛ inch (0.3175 cm.) to 5/16 inch (0.7938 cm.) in diameter, by approximately 12 inches (30.48 cm.) in length, likewise depending upon the class of welding. For welding in wrought iron and steel, nothing can excel the genuine Norway or Swedish iron.

Electric welding, more especially when effected by the Benardos or carbon-arc process, requires, on account of the brightness of the arc, to be done in an inclosure. The operator must also be thoroughly well covered over the body, head, eyes and hands. Exposure to the arc rays, even if of brief duration, causes an irritation or inflammation closely resembling sunburn, but fortunately with no more serious consequences.

Hoods may be made of canvas, sheet metal or insulating material and should be thoroughly well ventilated. The clothing is sufficient protection for the body and gloves for the hands. As the eyes are so much more sensitive, even greater care should be taken in their protection.

FLUXES THAT ARE USED IN WELDING.

A diversity of opinion seems to exist as to the value of fluxes, and many operators have their own secret formulas. It may be stated, however, that no flux is needed for welding either wrought iron or steel, the field to which arc welding has thus far proved to be particularly adapted.

For welding wrought iron or steel, various filling materials may be used, among them being Norway or Swedish iron rods, boiler iron, scrap, bits of steel castings, etc.; for cast or malleable iron, Norway or Swedish iron rods or special cast-iron rods with a high percentage of silicon.

In joining two pieces of any but the thinnest sheet metal, the abutting edges should first be chamfered at an angle of about forty-five degrees, either from one or both sides, depending upon the thickness of the material, and the chamfer should extend entirely through the thickness of the plates.

As soon as the weld has been completed and while still at a white heat, it should be briskly hammered to give the metal a finer grain. All oxide and other impurities should, of course, be kept out of the weld.

Heating the metal preparatory to welding is always beneficial in preventing and relieving strains, but is not resorted to except in special cases. A temporary furnace, especially for large work, is easily made of firebrick when needed. Annealing is also desirable at times, but, like preheating, is not done except when necessary.

THE WELDING OF BOILER TUBES.

The welding of flues in locomotive boilers offers a most fertile field for the Slavianoff, or metal-electrode, process. Several different methods of preparing the flues and the flue-sheet preparatory to welding are in use,

one of the most promising being that developed by O. C. Wright, assistant engineer of motive power of the Pennsylvania Lines West. Referring to Fig. 1, the flue sheet is first countersunk around each flue hole for a width of ¼ inch (0.635 cm.) to ⅜ inch (0.9525 cm.) and to a depth necessary to give a surface of clean metal and a feather edge at the flue hole. The flues are next driven into position in much the usual manner, except that no copper ferrules are required nor is it necessary to swell the flues just back of the flue-sheet; further, the flues should project above the countersunk surface of the flue-sheet not to exceed 1/16 inch (0.1588 cm.) at any point, and the flue rim should be cut away, both top and bottom, through an arc of about ½ inch (1.27 cm.) as shown. In making a weld, current of approximately 90 ampères at 18 volts across the arc is employed, the welding commencing at the bottom and working around to the top for each half. Care should be taken to play the arc on the flue sheet immediately adjacent to the flue, rather than on the flue directly, and it is also preferable so to proportion the length of the welding rod, which should be of 5/32 inch (0.397 cm.) diameter Norway iron, that each half of the weld will be made with a single electrode and, if possible, with one application of the arc. No flux is necessary. Including the cutting away of the rim, about twelve flues can be welded per hour.

Flues welded in this manner have been tested to destruction, withstanding a crushing force of over 30 tons (27,200 kg.) along the axis of the flue, at which pressure the flue itself failed; this as against 17 tons (15,400 kg.), at which pressure flues applied in the usual manner are found to fail.

Both of the electric-arc processes are particularly well adapted to the welding of wrought iron and steel of various kinds, and while certain other metals, including cast iron, can also be welded, the work is not, generally speaking, done to advantage over some of the other processes. Welds in cast iron will prove quite variable when made by the average operator, being frequently glass-hard (though this may not matter where no machining is done) and uncertain as to strength, so that at present it would be unwise to make any pronounced claims for arc-welding in this connection. With malleable iron the results obtained are even more variable than with cast iron. In the welding of the latter, good results are obtained by using the carbon-electrode process with a suitable cast-iron filling rod, first preheating the metal to white heat, making the weld while in this condition and then cooling gradually.

Blacksmith versus Carbon-Arc-Welded Rings: The following figures have been obtained in the works of the Westinghouse Electric and Manufacturing Co., East Pittsburgh, Penn.:

Sections.	Smith Welds.*	Arc Welds.*
1 x1½ inches	$0.59	$0.51
1¼x1½ inches	0.66	0.30
1½x2 inches	1.13	0.45
1¾x2½ inches	1.25	0.45
2 x6 inches	3.05	0.85

*Labor costs only included.

The following figures were supplied by one of the large American railroads, being taken from jobs at various times and compared with the cost of restoring the apparatus to service by previous methods, either by replacement or repair. The arc-welding costs were based on an hourly power cost of 51 cents for the carbon arc and 17 cents for the metal arc, together with the cost of direct labor and an overhead charge of 40 per cent. The power costs are somewhat higher than those obtained in average work.

COSTS OF RAILROAD REPAIR WORK.

Operation.	Cost of Welding.	Cost of Replacement or Repair by Former Methods.
Welding tender draft casting	$1.11	$18.31*
Welding tender center draft casting	4.36	19.06*
Welding tender draft arm	1.11	19.08*
Plugging 51 holes in expansion plate, hole 1 in. dia. by ½ in. deep	2.75	10.15
Repairing mud ring	6.50	34.57†
Building up flat spots on locomotive drivers	.40	225.00‡
Building up four piston valve flanges	9.52	24.20*
Repairing mud rings	5.57	32.70†
Plugging four holes 2 in. dia. by 4 in. thick	1.16	4.44
Cutting four 6 in. holes in tender deck sheets ½ in. thick	1.08	8.35
Plugging holes in two driving-box cellars	.55	2.55
Welding eccentric strap, broken through neck	1.08	41.28*
Building up jaws of two pedestal caps	3.49	10.00
Welding one driving box for type K-2 locomotive, broken through crown	5.75	39.31*
Welding frame stiffener on locomotive	3.29	14.73
Welding main rod, broken through end	6.35	70.49*
Repairing firebox	134.89	869.58§
Welding two spokes in trailer-wheel center	7.72	68.05*
Welding two spokes in driving-wheel center	7.98	99.98*
Welding three spokes in driving-wheel center	11.20	126.60*
Welding two spokes in driving-wheel center	5.13	112.09*
Welding cracks in bulkhead in tender crank	2.33	8.00
Welding cracks in side and door sheets of firebox	4.23	24.35†
Welding bridge in flue-sheets	2.88	20.12†
Welding cracks in side sheets	26.15	31.79
Repairing air drum	2.83	12.64*
Welding guide yoke	1.68	16.15†

*New parts required. †Repair. ‡Estimated cost to turn down all drivers as would otherwise be required. This would mean also the loss of at least one year's wear on the tires. §New firebox required.

The Thomson Electric Welding Co., which was the pioneer in this process and which has therefore had the greatest experience in this field, presents the following as covering the metals and alloys that can be welded:

Metals: Wrought iron. Cast iron. Wrought copper. Lead. Tin. Zinc. Antimony. Cobalt. Nickel. Bismuth. Aluminum. Silver. Platinum. Gold (pure). Manganese.

Alloys: Various grades of tool steel. Various grades of mild steel. Steel castings. Chrome steel. Mushet steel. Stubs steel. Crescent steel. Bessemer steel. Nickel steel. Wrought brass. Gun metal. Brass composition. Fuse metal. Type metal. Solder metal. German silver. Aluminum alloyed with iron. Aluminum brass. Aluminum bronze. Phosphor bronze. Silicon bronze. Coin silver. Various grades gold.

Combinations: Copper to brass. Copper to german silver. Copper to gold. Copper to silver. Brass to wrought iron. Tin to zinc. Tin to brass. Brass to german silver. Brass to platinum. Brass to tin. Brass to mild steel. Wrought iron to cast steel. Wrought iron to mild silver. Steel to platinum. Wrought iron to tool steel. Gold to german silver. Gold to silver. Gold to platinum. Silver to platinum. Wrought iron to mushet steel. Wrought iron to stubs steel. Wrought iron to crescent steel. Wrought iron to cast brass. Wrought iron to german silver.

Wrought iron to nickel. Tin to lead. Mild steel to tool steel. Nickel steel to machine steel.

ELECTRO-PERCUSSIVE METHOD OF WELDING.

As electro-percussive welding is of more recent origin than any of the other methods, less is known of its ultimate possibilities, more especially with reference to its adaptability.

It will weld any two metals, whether alike or unlike as to their composition, melting point, conductance—in fact, quite regardless of their characteristics, though such combinations as silver and tin or aluminum and tin, iron or lead will of course not be permanent any more than they will be when welded by any other method, since their alloys disintegrate in time. Thus far this method has been largely confined to the welding of small wires or the equivalent, aluminum to aluminum or copper, platinum to nickel or copper, thermo-couples, spring steel, etc.

The principle involved is that of percussive contact between two metals with a condenser discharge occurring at the instant of such contact. The various items involved include voltage, velocity and force of impact.

At the instant of contact, the condenser current is so heavy that the ends of the abutting metals are vaporized by the explosive discharge; and, owing to the blow delivered by the falling mass, are forged together. The heat produced, though intense, is developed so quickly that it is confined to the extreme surfaces of the metals and therefore there is no opportunity for unequal heating with its possible deleterious effect on the welded materials.—"Power."

HEAT TREATMENT OF STEEL.

The shipbuilding industry and that of marine engineering are largely concerned with the production of iron and steel, and anything that is written or done by those who are competent to deal with such matters in the direction of improving material belonging to either class is of interest to those engaged in either branch of ship construction. A paper that has been widely discussed in United States technical circles is that which was read before a meeting of the American Society of Mechanical Engineers by Mr. Robert R. Abbott, and which dealt with the heat treatment of modern steels. In broadly summarizing the facts connected with this subject the author remarked that the process of heat treatment consists practically in transforming an alloy from a mechanical mixture of two substances into a homogenous solution of a single substance. This very fact gives to a heat-treated steel one of its most valuable advantages over untreated steel, namely, it ability to withstand repeated shocks or vibrations or, as it is popularly expressed, fatigue-resisting power. The reason is obvious; an alloy composed of two dissimilar substances has mechanical boundaries between them. This fact is a source of weakness under repeated stresses, the reasons for which are more or less complicated. An untreated steel contains these boundaries, but a correctly heat-treated steel does not. A steel with a tensile strength of, say, 60,000 pounds per square inch, if subjected to repeated reversals of stress of any magnitude will finally break after a definite number of these reversals, when heat treatment considerably improves the performance of the steel in this direction.—"Shipbuilding and Shipping Record."

MODERN MACHINE TOOLS; RECENT DEVELOPMENTS. By L. O. ALFORD. "Pages Engineering Weekly," November 12, 1915.

FATIGUE OF COPPER ALLOYS. By ERNEST JONSON. "The Brass Weekly," December, 1915.

OXY-ACETYLENE WELDING PRACTICE. By S. W. MILLER. Treats of the Metallurgy of Iron and its Relation to Welding; Difference between Cast Iron, Wrought Iron and Steel; Welding of Steel; Burning of Steel; Method for Welding Steel; Heat Treatment of Welded Steel, etc. "Machinery," December, 1915.

SUBMARINES.

PRESENT CONDITION OF THE SUBMARINE.*

BY MAX A. LAUBEUF, LATE CHIEF CONSTRUCTOR, FRENCH NAVY.

The Two Types, Submarine and Submersible.—It has often been said, and wrongly, that the submersible is distinguished from the submarine of earlier type because it has two motors, one for running on the surface and the other for running immersed. It is true that in France the *Narval* was the first boat destined for underwater navigation, and provided with two motors. The submarines which had preceded had but a single motor, a motor of the electric type with storage battery, with a consequent serious reduction in the radius of action. But two motors may be placed on the submarine of earlier type, as has been done. This is what was done particularly in the United States by Mr. John Holland, in 1893, on the *Plunger* (a steam engine and an electric motor), and, in 1897, on the *Holland* (a gasoline engine and an electric motor).

The differences between the submarine of earlier type, or the submarine pure and simple, and the submersible are rather differences in mode of construction, reserve of buoyancy and form of body.

1. *Mode of Construction* (see sections).—With the submarine pure and simple the water-ballast tanks (or tanks intended to submerge and trim the boat in the submerged condition) are placed inside the boat, which has circular cross-sections. With the submersible type, these reservoirs, of much greater volume, are placed outside the hull proper. There results an entirely different mode of construction. The submersible has a double shell. The inner skin, which must resist the pressure of the water, is made of thick plates. The shape of the sections is not circular, but elliptical. The outer skin, which is only subject to the pressure of the water when the ballast-tanks are empty—that is to say, when running on the surface—is made of very light plates, as in torpedo-boat construction. The double hull may be complete, as in the case of the *Narval*, or partial, as in later examples of the same type.

2. *Reserve Buoyancy.*—If the term "reserve buoyancy" is given to the emerged volume of a boat navigating on the surface, this volume, in the case of ordinary ships, is approximately equal to the immersed volume. In many cases it is even greater. If we denote by the term "coefficient of reserve buoyancy" the ratio of the reserve buoyancy to the total volume of the ship completely immersed, this ratio, which is at least 50 per cent. for ordinary seagoing ships, was no more than 3 to 7 per cent. for the earliest submarines, and scarcely exceeds 12 or 13 per cent. for the submarine type pure and simple. It is much greater for the submersible type. The boats built according to my designs have 27 to 33 per cent. The *Narval* had even 41 per cent.

It results from this important difference that the submarine pure and simple, low in the water and with small reserve buoyancy, readily drives into the waves, and that on the open sea it soon becomes necessary to withdraw the crew from the exterior and to close all openings. The boat then navigates on the surface under the same conditions as if im-

*Abstract from a paper presented at the International Engineering Congress, 1915, in San Francisco, Cal., September 20 to 25, 1915.

mersed. The submersible, on the other hand, navigates under such conditions exactly like an ordinary ship. The difference regarding comfort of living and fatigue of the crew is considerable.

The French maneuvers of 1902 with submarines of the *Morse* type and submersibles of the *Sirène* type, of 150 tons; the comparative trials, in 1905, between the submarine "*Z*" and the submersible *Aigrette*, of 180 tons; and, finally, the maneuvers of 1909 between submarines of the *Emeraude* type and submersibles of the *Pluviose* type, of 400 tons, have definitely demonstrated that the submersible type possesses nautical qualities far superior to the other.

3. *Form.*—A difference no less considerable exists in regard to the external form. The submarine pure and simple is distinguished by circular sections, and for many years it has been given a pointed form at both ends, similar in general to the cigar, the classical form for boats of this type. It was the general opinion, furthermore, that this form would alone permit of diving under good control. When, in 1897, the design of the *Narval* was presented, in which the external form was exactly similar to that of a torpedo boat, many persons (and especially officers acquainted through experience with underwater navigation) expressed the opinion that a boat thus formed could never dive satisfactorily. Experience alone could determine the question. It has proved, in effect, that the submersible type is as well suited to diving as the other.

The most serious objection to the submersible type, at the beginning, was that, in order to pass from the condition of surface navigation to that of underwater navigation, a change had to be made from the heat prime mover to the electric motor, and then a considerable volume of water had to be taken in between the double shell of the hull in order to annul the very considerable reserve of buoyancy. But during the time thus required the boat was in a critical condition, permitting destruction by the ship of an enemy without the chance of escape.

In point of fact, on the *Narval*, at the beginning of the trials, the time required for immersing reached 28 minutes and the objection noted above was fully justified. But on this same boat, with the progressive training of the crew and after some changes, this time was reduced successively to 20, then to 15, and finally to 12 minutes. On the *Sirène* type of 1900, which followed, the time required from surface navigation to underwater navigation was reduced to 8 minutes, on the *Aigrette* type of 1902 to 6 minutes, and, finally, on the submersibles of the *Pluviose* type of 1905 to less than 5 minutes. The objection formerly made has, therefore, no further significance; a period of 5 minutes answers all military requirements.

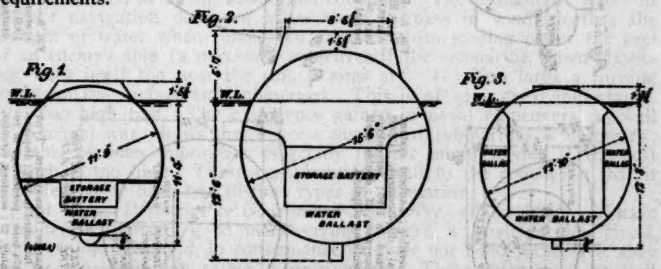


Fig. 1.—Submarine: Holland system. Type, Adder; launched in 1900; displacement, 108/122 tons; percentage of reserve buoyancy, 11.5; sections, circular.

Fig. 2.—Submarine: Electric Boat Company's system. Type K; launched in 1913; displacement, 390/520 tons; percentage of reserve buoyancy, 25; sections, circular.

Fig. 3.—French submarine: Maugas system. Type, Farfadet; displacement, 194/300 tons; launched in 1901; type, Emeraude; displacement, 398/482 tons; launched in 1906; percentage of reserve buoyancy, 7 to 8; sections, circular.

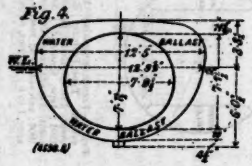


FIG. 4.—Submersible: Laubeuf system. Type, Narval; displacement, 117/202 tons; launched in 1899; percentage of reserve buoyancy, 42; double shell complete; sections, elliptical.

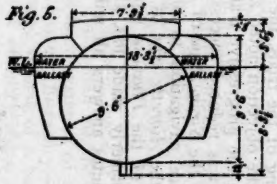


FIG. 5.—Submersible: Laubeuf system. Type, Sirène; launched in 1901; type, Algrette; launched in 1904; percentage of reserve buoyancy, 28 to 30; partial double shell: sections, circular.

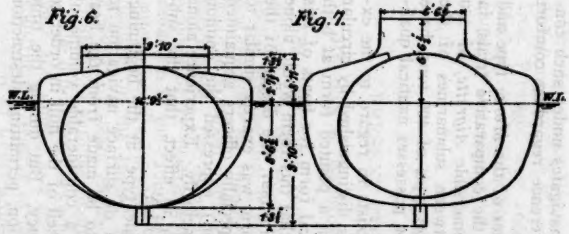


FIG. 6. Submersible: Laubeuf system. Types launched in 1907-1910; percentage of reserve buoyancy, 25 to 30; partial double shell; sections, elliptical.

FIG. 7. Submersible: Laubeuf system. Latest types launched in 1911-1914; percentage of reserve buoyancy, 23; partial double shell; sections, elliptical.

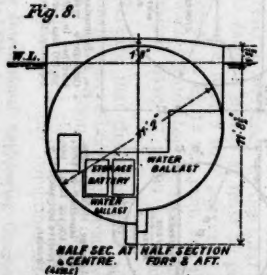


FIG. 8.—Submarine: Lake system. Type Protector; launched in 1902; displacement, 136/174 tons; percentage of reserve buoyancy, 22; sections, circular.

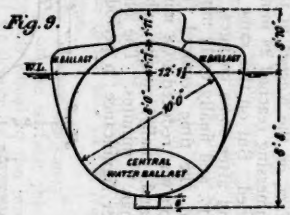


FIG. 9.—Submersible: Germania-Krupp system. Type, Kobben; launched in 1908; displacement, 202/255 tons; percentage of reserve buoyancy, 21; partial double shell; sections, circular.

FIG. 10.—Submersible: Laurenti system. Type, Foca; launched in 1908; displacement, 186/265 tons (with upper W.B.); percentage of reserve buoyancy, 30; partial double shell.

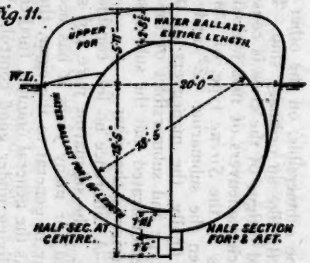


FIG. 11.—Submersible: large displacement: Laurenti system. Under design according to Berling, 1912; complete double shell for 20 per cent. of length; partial double shell at ends; sections, circular.

The diagrams, Figs. 1 to 11, in connection with the present paper, show plainly the difference of construction between the "submarine" and the "submersible." In 1900 France was the only naval military Power possessing submersibles. All the other naval Powers were building only submarines of the earlier types. Since that time the submersible type has made great progress. In France comparative trials demonstrated the superiority of the submersible. Since 1907 only submarine craft of this type have been built, and the submarines pure and simple of earlier construction, to the number of thirty-nine, have been gradually retired, There remained on January 1, 1915, no more than seven of these in service. The submersible type has been adopted, exclusively, by Germany, Italy, Greece, Sweden, Norway, Portugal and Brazil. Other naval Powers are developing experimentally the submersible and the submarine—England, United States, Russia and Japan. Finally, the submarine type pure and simple is undergoing modification, and is approaching, little by little, to the submersible. For example, the American firm which has built, either directly or through its licensees, the greatest number of submarines pure and simple, the Electric Boat Company, has long built boats of small reserve of buoyancy. This company is now turning to the submersible type.* Finally, this firm, which until 1913 built submarines with a single shell and with water-ballast tanks inside, is now building a new type, "*M*," with large reserve buoyancy and with a partial double shell. This is a marked change towards the submersible type, and I believe that I am justified in a feeling of deep satisfaction at the success of the ideas for which I have stood during the past eighteen years.

*See "Engineering," November 17, 1911.

Increase in Displacement.—Submarines have not escaped the tendency, which makes itself so strongly felt, towards increase in displacement. It may be asked if, in their case, this tendency is quite justified. In passing from 15,000 to 25,000 tons or more, armor-plated ships have had their speed, and particularly their offensive power, increased. The same remark applies to all ships whose principal armament is the gun. But ships which have the torpedo as the sole or preponderating offensive armament do not find such marked advantage in increase of displacement.

For submarines the increase of displacement provides a rather feeble increase of offensive power, but to obtain the greatly augmented speeds it is necessary to pass to large displacements. On the other hand, it may be said that a large displacement offers serious drawbacks for a submarine, as follows: (*a*) Greater difficulties of evolution while submerged. (*b*) Too great a draught of water in the condition of navigation on the surface, as well as in the submerged condition. The draught of water in surface navigation does not allow them to pass in small depths; the draught of water when submerged prevents them passing under the keel of an enemy's ship (a necessary maneuver if the submarine, when attacking, finds itself too near the ship it aims at). (*c*) Too large a turning circle, on the surface and submerged. This is of very great importance. (*d*) Too high cost. The experience gained in naval maneuvers, as well as in actual war, shows that a great number of submarines is necessary; and this becomes impossible, especially for the minor navies, if the cost of each is too high. These considerations lead to the conclusion that it is preferable to have two distinct types of submarine.

First Type: Defensive or Coastguard Submarine.—These boats will have a moderate tonnage, good maneuvering powers, a small turning circle, moderate draught, and, in consequence of their not being expensive, they can be constructed in rather large numbers. These ships must be well armed, but they need not realize very great speeds nor have a very large radius of action when navigating on the surface. This class of submarines exists almost everywhere. It is represented in France by the *Pluviose*, *Brumaire* and *Clorinde* types of 400 to 412 tons on the surface, 550 to 560 tons when submerged; in the United States by the *C*, *D*, *E*, *F*, *G*, *H*,

K classes, of 238 to 390 tons (surface), 273 to 520 tons (submerged); in Germany and Austria by the type *U 2*, of Germania-Krupp, of 240 to 300 tons; in Italy by the type *Foca*, of Laurenti, of 245 to 300 tons.

These latter types have too low a displacement. This class of submarine must, as a matter of fact, not only defend the coasts, but also take the offensive within a fairly large radius from the base of operations. They must therefore have good armament, excellent nautical qualities and good quarters for the crew, all of which is very difficult to obtain with a small tonnage. Accordingly, they must have a high reserve buoyancy. In French submersibles of the *Pluviose* and *Brumaire* classes, laid down in 1905–7, the reserve buoyancy is 150 tons, or 27 per cent. of the submerged displacement. In the United States this percentage is increasing more and more from the "*C*" class (14 per cent.) to the "*K*" class (25 per cent.). The good seagoing qualities of submersibles with large reserve buoyancy have been proved by a great number of journeys of long duration made by ships constructed after my plans.†

†*Ibid.*, July 9, 1915, page 29.

Second Type: Squadron Submarine.—The second type of submarine, which it is now sought to realize, is the high-seas or squadron submarine. The latter type, intended to accompany the fighting ships, to sail with a squadron in any weather, to take part in naval battles, must in future replace the destroyer type. The enumeration of the operations in which it is to take part shows that it is absolutely necessary for this boat to possess high speed on the surface and good speed when submerged; a large radius of action, especially on the surface; great offensive power, good nautical qualities, and excellent living quarters. These are difficult conditions to fulfil, and can only be realized with a very considerable displacement.

It may be said at once that ships of intermediate tonnage, ranging between the coastguard submarine and the squadron submarine, constructed or in course of construction, are useless; they are too large as coastguard submarines and too small as high-seas submarines. I include in the number of these ships, which should never have been put down, all the submarines of 600 to 700 tons on the surface, constructed or in course of construction. All these ships represent but half measures. They can not accomplish much more than coastguard submersibles of 400 to 500 tons on the surface; they possess too low speed and inadequate nautical qualities to accompany a squadron on the high seas. They are likely to disappear before the submarine with great speed and of much greater tonnage.

Three types of submarine, only, approach closely at the present time to the true conception of the squadron submarine. The French Navy is in the lead. The *Gustave Zédé*, ordered in 1910, launched on May 20, 1913, is now commissioned. The *Nereide*, ordered in 1911, launched on May 9, 1914, is completing. They have 800 tons on the surface, 1,100 tons submerged. Two others, a little greater, were ordered in 1913 and five in 1914. They have 835 to 1,200 tons. In the United States, a new type, "*M*," of the Electric Boat Company, is projected. The displacements are understood to be 1,000 to 1,500 tons.

If my information is correct, the program of squadron submarines in France, Great Britain, and the United States is similar—19 to 20 knots on the surface, 11 to 12 knots submerged. Will these types of submarines adequately represent squadron submersibles? I answer, without hesitation: "No; their speeds on the water and under the water are still insufficient." To be able to accompany a squadron in any condition of the sea and take an effective part in a naval battle, it is necessary to have a maximum speed on the surface at least equal to that of the modern armored ships—viz: 22 to 23 knots. I say "at least" because, as is well known, small ships lose much more speed than large ones in a heavy sea, and a submarine, even if it had 1,000 tons displacement on the surface, is still but

a small ship on the open sea. It is necessary, also, for the squadron submarine to have a speed, when submerged for battle, at least equal to the current speed of evolution of the armored ships of the line at the *Narval* had even 41 per cent.
time of the battle. And here a speed of 15 knots appears to me as an absolute minimum.

Consequently, I say that, as long as submarine boats cannot attain speeds of 23 knots on the surface and 15 knots when submerged, they will be insufficient to fill the rôle of the squadron submarine, and they will only impede armored ships.

What, now, are the reasons which prevent the realization of these speeds at the present time? (1) It has not yet been possible to construct very powerful internal-combustion motors, which would at the same time have a relatively small weight and size. If the immense progress accomplished in the last ten years in the construction of internal-combustion motors with four and two cycles is considered, one is led to believe that within a few years we shall have the necessary motors. (2) The second point, and the more weighty one, is the double motor. Besides the internal-combustion engine, the submarine has an electric *ensemble* consisting of motor and a battery of extremely heavy and cumbersome accumulators. We are under a great debt to the electric-storage battery. Without this, without the realization of a source of energy which does not present variation in weight when delivering the accumulated energy, which absorbs neither air nor oxygen and discharges but few noxious gases, submarine navigation would perhaps not exist at all.

Today, showing ingratitude, we should like to banish the electric battery from the submarine and to place in these boats one motor only, supplying the necessary power for propulsion on the surface as well as when submerged. Many attempts have been made in this direction, some with a single internal-combustion motor, others with a single steam engine, but, up to the present time, no boat with a single motor has solved the very difficult problem which is before us. Searchers and inventors have, therefore, an open field before them. When they have given us the powerful, sure motor, with relatively small weight and size, which is so much desired by the builders of the submarine, the program of squadron submarines set out above can be realized, and the destroyer may disappear from modern fleets. Till then we can only have, presumably, inaccurate approximations to a solution of the problem of the squadron submarine and with very great displacement.

To sum up, the present tendencies of the constitution of submarine flotillas appear to be the following:

1. *Coast-Guard Submarines.*—Coast-guard submarines of moderate displacement, 350 to 500 tons on the surface, 500 to 750 tons submerged; well armed (for example, two inner torpedo tubes, and outer tubes or outer torpedo-launching equipment, with six or eight torpedoes on board); with suitable speeds (14 to 16 knots on the surface, 9 to 10 knots when submerged for battle); having a radius of action reasonably fixed in accordance with their probable operations and the geographical conditions of the country which they are to protect; having good living quarters; and, finally, with good nautical qualities, in order that they may be able to take the offensive within a fairly large radius, and this, in particular, implies a high reserve buoyancy.

2. *Squadron Submarines.*—Squadron submarines having a great displacement (without going too far, with a view to remaining within dimensions that do not create too serious difficulties of evolution or of use; for example, 1,200 tons when submerged); having high speed (23 knots, at least, on the surface, 15 knots, at least, when submerged), with a powerful armament (for example, eight inner torpedo tubes and sixteen torpedoes on board); a very large radius of action on the surface; excellent quarters; and first-class nautical qualities, implying a high reserve buoyancy.

The conclusions which precede represent the ideas which I have defended for many years. I have had the satisfaction of seeing these ideas adopted by the French Navy, then, in 1913, by the English Admiralty. Finally, since writing the preceding lines, I have received a copy of the report of the United States General Naval Board, of which Admiral Dewey is President, and which appeared in December, 1914. This report recommends the construction of two types of submarines, one for the high seas and the other for coast defence, and asks the building of three boats of the first type and sixteen of the second, adding that no intermediate type seems to be called for.

The submarine has already taken an important place in the composition of war fleets. This importance can only increase in the future. The war of 1914–15 marks an important date in the evolution of naval material.—"Engineering."

THE SUBMARINE OF TOMORROW.*

By L. Y. Spear, Member.

[The author in the first part of the paper dealt with "the submarine of today," giving particulars of existing vessels, but the second part, dealing with "the submarine of tomorrow," is most interesting, and is here reproduced.]

Enough has already been said to indicate that up to displacements of 750 tons and speeds of 17 and 11 knots, the designs of "the submarines of tomorrow" will be fixed, not by engineering limitations (as in the past), but by military (including financial) considerations.

Any opinion as to whether the very small harbor or local-defence type is destined to be perpetuated lies outside the scope of this paper, but it may not be out of order to venture a guess in the negative. At any rate, if this type is found in future naval programs, that fact will constitute a marked exception to the ordinary history of naval development. On the other hand, the coast-defence and offensive types are sure of perpetuation. In fact, their value is now so fully recognized as to make it certain that present and future naval programs must include such boats in large and increasing numbers. It is also obvious from what has been said above that no standardization as to the size of future submarines is to be expected, since the most effective displacements most necessarily vary with the different conditions met in different cases. This being so, limitations of space here make it necessary to confine our further inquiry into the subject to the case of the United States.

At the present time it appears to be the policy of the United States to employ the submarine extensively for coast-defence purposes and also to construct it for offensive purposes, either in conjunction with, or independent of, the battleship fleet. In view of the great distances which separate us from the most of our probable enemies, and certain special characteristics necessary for the offensive type, it appears safe to conclude that in the immediate future at least two quite separate types will be employed, one of which we shall designate as the "defensive submarine," and the other as the "fleet submarine." While these two types may very well present marked differences in characteristics, nevertheless there are certain general conditions which in the nature of the case must apply to both, and therefore, before attempting to outline the probable future of each type, it will be well to give consideration to these basic conditions.

The outstanding features characterizing submarine warfare are the following:

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*Paper read at the twenty-third general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 18 and 19, 1915.

(*a*) The submerged speed is necessarily inferior to that of the principal objective, viz: the enemy's capital ships.

(*b*) The protection of the submarine while engaged is independent of its displacement, except to the extent that increased dimensions impair handiness.

(*c*) The torpedo power is not proportional to displacement.

All of these considerations emphasize the importance of numbers. Financially speaking, however, large numbers involve limited displacement, and hence (within limits) decreased efficiency of the unit. How is this basic difficulty to be adjusted? At first glance the question seems too involved for any precise treatment, but on careful examination we find a reasonable basis for a solution in the fact that physical reasons, for the present at least, place a limit on the submerged speed of from 10 to 12 knots. As the chances for a successful attack on a moving target are so vitally affected by the submerged speed, it seems reasonable to adopt the practical maximum as a *sine qua non*, and from that basis build up the design. As the torpedo is the sole reason for the existence of the submarine, it should not be too difficult to fix some irreducible minimum for the number of torpedo tubes, which in this case may be put at four. The third absolutely essential quality is the ability to pass from the surface to the submerged condition in a very short period of time. A precise limit as to the permissible period of time is difficult to set, but, taking everything into account, we may here regard three minutes as the maximum permissible allowance. By eliminating all frills and non-essentials it is possible today to secure these three primary qualities in a well-balanced design of about 250 tons surface displacement. Starting with this as the lower limit, let us now consider the additional factors, partly military and partly technical, which must determine the final solution.

Taking up first the remaining primary military features—viz: submerged radius, surface speed and surface radius—we are confronted first by the fact that on account of the necessity for separate power plants, other things being equal, an increase in any one of these qualities can only be obtained by a sacrifice in one or both of the others. Keeping in mind the fact that a submarine on the surface has only a minor military value, it is natural first to set some reasonable minimum on the submerged radius. While opinions may well differ on this subject, the author holds the view that a radius of at least 100 miles at a speed of about 5 knots is necessary. This requirement by itself will not raise the minimum displacement figure above mentioned—viz: 250 tons—and will permit of a surface speed of from 11 to 12 knots, with a fuel supply corresponding to a radius of 1,000 miles.

Leaving aside for later consideration the question of the "effective" radius of action, which, in fact, involves much more than the mere fuel supply, let us briefly consider the effect on displacement of increasing the surface speed only. We have only to remember the intimate relation of length to speed to see at once that while keeping the submerged qualities constant, increases in surface speed are expensive in displacement. Thus to increase the speed of the above primary design by two knots, it is necessary to increase the length from 115 feet to 145 feet, and to increase the surface displacement from 250 to 335 tons, and an increase to 20 knots surface speed would involve a length of about 260 feet and a displacement of about 1,000 tons.

Increases in radius of action, in so far as the mere ability to carry the necessary fuel is concerned, do not make such great demands on displacement; but when all of the factors which in practice fix the "effective" radius are taken into account, the demands on displacement become astonishingly great. The "effective" radius is, in fact, that distance which the boat can run on the surface, taking the weather and sea as it finds them, and still arrive at destination with the personnel in efficient physical con-

dition for submerged work. In other words, habitability and seaworthiness from a habitability point of view are vital factors, so vital, in fact, as to warrant some detailed examination here.

In the case of the 250-ton primary design above referred to about four days would be required to run out the fuel radius. Good practice, then, would require fresh water and provisions for a week, which might be taken as the limit of time for such a boat to be absent from its base and dependent entirely upon its own resources. If good weather or operation in sheltered water could always be counted on, the complement of such a boat could be organized in a watch in two, and be thus reduced to a number for which it would be possible to provide living accommodation of an elementary character. Now let us assume that an effective radius of 3,000 miles is required. Obviously, such a radius involves work in the open sea with a possible period of independent operation of three weeks. Here a watch in three is at once indicated, with an increase in the number of the crew of about 50 per cent. Moreover, the living accommodation must be much more elaborate, and the fresh-water and provision supply must be four and one-half times as great. This, however, does not tell the whole story, since reasonably comfortable conditions for surface navigation for extended periods in rough water, or very cold weather, involve alterations in the design, which, if not compensated for by additional displacement, would reduce the submerged speed and radius. For instance, the percentage of surface buoyancy which would be adequate with a smaller radius would, other things being equal, not be adequate in the case of the larger radius. An increase in this, without increase in displacement, would involve a decrease in submerged speed and radius. Again, even if the supposedly fixed submerged qualities were sacrificed to increase the habitability, we must remember that absolute displacement is in itself a large factor. In other words, having in view the weather conditions which must be met, there is probably some minimum limit of surface displacement, even when substantial sacrifices of other qualities are made in favor of habitability.

Having thus briefly outlined the design problems flowing from the purely military considerations, let us turn now to the engineering questions involved. Assuming certain speeds, radii and armament as rigid requirements, it is obvious that the displacement must vary according to the type of machinery installed and the amount of attention paid in the design to questions of access and reliability. The most important items in this connection are, of course, the main engines and the storage batteries. In each case reliability and long life are in conflict with compactness and light weight. In these important respects there is as yet no uniformity of practice in the different navies of the world. In certain Continental countries, notably Germany, the extreme of lightness and compactness is sought, accessibility is sacrificed, and a very highly-trained personnel is relied on for reliability. The wide range of practice is illustrated by the fact that the weight in pounds per horsepower of the main engines of the single-acting Diesel type almost universally used varies from 50 to 100. In the case of storage batteries, the weight per motor horsepower at the one-hour rate of discharge varies from 131 pounds to 253 pounds. The smaller figure represents the extreme case, where the durability of the battery is sacrificed to capacity. The heavier of the two batteries is, in fact, expected to have a service life at least three times as great as the lighter. Quite aside from the type of machinery adopted, the matter of access and roominess in general is of great importance, and, finally, we must, in addition, consider the demands made by safety features, including elaborate bulkhead subdivision, as well as the installation of special fittings and appliances, such as submarine signals and the like.

To illustrate the aggregate effect of such considerations, we may compare a comparatively simple design of date of 1910 with a very recent one. The early design has the following characteristics:

Surface speed, knots	14
Surface fuel radius, miles	2,490
Submerged speed, knots	11½
Submerged radius at 5 knots, miles	110
Armament	4 torpedo tubes and 4 torpedoes.

The corresponding data for the later design are as follows:

Surface speed, knots	14
Surface radius, miles	3,150
Submerged speed, knots	10½
Submerged radius at 5 knots, miles	75
Armament	4 tubes, 8 torpedoes and 1 3-inch gun.

It will be noted that the newer design provides for slightly greater armament, with an increase in the surface radius. On the other hand, this has been compensated for by a decrease in the submerged speed and radius. Notwithstanding this reduction in submerged qualities, the surface displacement is increased 60 per cent. In this particular case there is no material difference in the weight and space per horsepower of the main engines, and the very large increase in displacement has nearly all been absorbed by habitability and safety features.

The substantial influence of the type of main engines employed may also be best illustrated by an example from actual practice. In the case in mind, on a surface displacement of 350 tons, a trial speed of about 14½ knots was obtained with comparatively light two-cycle engines developing 450 horsepower at 450 revolutions per minute. The engines in this case weighed about 56 pounds per horsepower. When these engines were replaced by a heavy type, weighing 86 pounds per horsepower and operating at 375 revolutions per minute, the trial speed was reduced to 13 knots.

Having thus briefly reviewed the main elements of the problem, we are now in a position to consider the probabilities with respect to the characteristics of the United States submarines of the near future. Let us consider, first, the question of the defence type. Adequate defence obviously requires the assembly of the requisite number of boats where and at the time needed. Broadly speaking, this may be accomplished in two different ways. The first method is in principle that of local defence, embodying the permanent stationing of an adequate number of boats of moderate radius in each locality where protection is desired. With adequate numbers, this method can certainly be relied upon to prevent blockades, bombardments or landing operations. The second method involves abandoning altogether the idea of assigning groups of submarines for the defence of certain areas, and using instead flotillas of larger boats, which, by reason of increased speed and sea-keeping qualities, could be concentrated when and where wanted. Such flotillas could also carry the line of defence much further from the coast, and in this way could coöperate more fully with the battle fleet. Should this second theory be adopted, it appears certain that there would arise an imperative demand for much higher surface speeds, which, of course, would involve marked increase in the displacement and cost of each unit. At the present writing the largest defence boats ordered by the United States are of about 500 tons surface displacement, with a speed of 14 knots. If the policy under discussion were to be adopted, it would perhaps be logical to pass at once to materially higher displacements and speeds, say 800 tons and 17 to 18 knots. Following the natural law of naval evolution, which seems to look toward the attainment of the maximum power in each unit, we would expect to see this displacement and speed increase from year to year. Carried to the extreme and logical end, this policy would eventually result in a merging of the defensive type into the offensive type. When that point is reached, if ever, our present units would probably be assigned

to local defence, and their numbers would be supplemented by new constructions of moderate displacement.

Having in view the strong differences of opinion on the subject which now exist in the Navy, the author does not anticipate the definite adoption at an early date of either method to the exclusion of the other. In fact, if the past is a safe guide, we may expect our Navy Department to keep "in the middle of the road," thus avoiding all sudden radical changes. It follows, then, that we may reasonably expect a compromise in the form of the construction of boats which, while larger than necessary or desirable for local defence purposes, can still perform that duty and at the same time possess the necessary habitability and the "effective" radius to enable them to keep the sea for long periods of time and to operate at a long distance from their base. Such a type would thus also be available for use according to the second method, and on the whole should be very efficient for such purposes, since, on account of the moderate displacement involved, it will be practicable to compensate for lack of high surface speed by an increase in numbers. In fact, if an increase in numbers be coupled with adequate scouting service, the final efficiency obtained from a given expenditure may be greater if the speed of the flotillas is moderate than if it be high.

As noted above, the necessity for sea-keeping qualities will in itself place some lower limit on the surface displacement. Actual experience, which is really the only safe guide, appears to fix this limit at about 350 tons. In this connection, it may be interesting to note some of the most notable achievements of submarines in the matter of long voyages in the open sea. Some years ago, the U. S. S. *Salmon*, now the *D-3*, of 288 tons surface displacement, made a successful trip from Boston to Bermuda and return, the journey involving a stretch of about 700 miles from port to port. This year a large number of submarines, of 350 tons displacement, have successfully crossed the Atlantic under their own power from Canadian waters, which performance, taking the number of boats and the length of the voyage into account, appears now to be a record. In the French Navy, the *Faraday*, of 400 tons surface displacement, has successfully made the passage from Rochefort to Toulon, a stretch of 1,730 miles. The voyages of the British "E" class from England to Australia, to the Dardanelles and to Russian Baltic waters, as well as the voyages of the German boats around the British Islands and also to the Dardanelles, are all very convincing as to sea-keeping qualities. In these last cases, the average surface displacement involved is about 700 tons.

In the special case under consideration the limit of permissible cost would fix the upper limit at, say, 550 tons. If we allow for the prevailing United States standards with respect to armament and the features affecting habitability, safety and access, as well as durability and reliability of machinery, a surface displacement of at least 500 tons is indicated. Assuming that the military conditions can be met by an effective cruising radius of 2,500 miles at 11 knots with normal fuel supply, it will be interesting to set down some possible solutions where the submerged speed and radius and the surface speed are the only variables. In all these solutions, armament, safety, habitability and roominess are the same, and the same character of machinery is contemplated, except where noted. The percentage of buoyancy is varied, as it should be, to meet the actual conditions as to surface speed and as to speed length ratio.

All things considered, the author believes that Design No. 1 presents the most valuable combination of qualities, but as this would involve a change in prevailing practice, it is perhaps more probable that the characteristics of United States coast-defence boats of the near future will more closely approach Design No. 2. Whatever may be the decision of our naval authorities as to the surface speed necessary, the author is firmly convinced of the relatively greater importance of the submerged

qualities for this type of boat, and, consequently, believes that battery capacity and submerged radius and speed should not be unduly sacrificed either in the interests of durability or first cost.

—	Design No. 1.	Design No. 2.	Design No. 3.
Displacement, surface, tons	500	500	500
Buoyancy, per cent. of above	20	24	28
Displacement, submerged, net tons	600	615	640
Surface radius at 11 knots, miles	2,500	2,500	2,500
Armament (all tubes internal)	4 tubes 8 torpedoes 1 3-in. gun	4 tubes 8 torpedoes 1 3-in. gun	4 tubes 8 torpedoes 1 3-in. gun
Surface speed, knots	13	14	15
Submerged speed, knots	11	11	10½
Submerged radius at 5 knots with*:			
Battery A	115	100	80
Battery B	132	115	92
Battery C	145	126	101

*Type "A" is a heavy, durable battery with positive plates of the Planté type. Type "B" involves the use of thinner plates of the pasted type with a probable sacrifice in durability of about 50 to 60 per cent. Type "C" is of special construction, equal to type A in life, but more expensive, so that its use would increase the first cost of the boat by from 4 to 5 per cent.

In the above discussion it has been assumed throughout that the type of machinery used, both for surface and submerged propulsion—viz: Diesel engines and electric motors—will be adhered to in the near future. The various substitutes which have been proposed, particularly with respect to the submerged power plant, are all as yet in the early experimental stage. Even should any of them achieve such success as to warrant the abandonment of prevailing types of machinery the results will not show in the defence "submarine of tomorrow," but rather in "those of the day after tomorrow," which are beyond the scope of this paper.

It remains still to consider the offensive or fleet submarine. If this type is to accompany the battle fleet a high surface speed is an absolute necessity, and with this must be combined a long effective radius. The question as to whether the fleet submarine really belongs among "the submarines of tomorrow" or not depends upon the surface speed which is considered adequate. If a speed of from 20 to 21 knots will suffice the answer is certainly "Yes"; but if 25 knots or more are required there is, to say the least, serious doubt with respect to the matter.

As the crux of this question lies in the matter of the surface power plant, it is necessary now to examine this subject in some detail.

At the present time almost all modern submarines depend for surface propulsion on Diesel engines, the only exceptions being a certain number of French boats equipped with steam engines. The Diesel engine is, in fact, the ideal motor for this purpose, on account of the following qualities:

(*a*) Compactness and moderate weight.
(*b*) Great economy in fuel consumption.
(*c*) Ability to start instantly.
(*d*) Ease in disposition of products of combustion.
(*e*) Freedom from excessive radiation or storage of heat.

In all respects, except quality (*a*), the Diesel engine is vastly superior to the steam engine, and hence is to be preferred.

The conditions under which any engine must work in a submarine are very unfavorable. The space and weight available are strictly limited, and the demand for speed often so insistent as to force the use of comparatively light-weight high-speed engines. Complete reliability of operation under such circumstances can only be attained after an adequate period of development. The submarine Diesel engine, while a newcomer in the field, has even now been developed to a point where, for moderate powers, entirely satisfactory reliability is obtained when extremes in the way of light weight and high speed are avoided. The best engineering talent in all countries is being devoted to the development of these engines, which is, in fact, making rapid strides. Up to the present time the maximum power per unit which is actually in successful service is about 1,200 horsepower, but considerably larger units are under construction. As is natural, owing to the short time in which this type of engine has been under development, uniformity of practice has not been reached. In Germany and in other Continental European countries, except France and Russia, very high-speed, light-weight engines are most in favor, with the two-cycle type generally predominating over the four-cycle type, although many of the German boats are equipped with four-cycle engines constructed by the Augsburg Works, the others being equipped with two-cycle engines constructed by the Krupp and Nuremberg Works. In France and Russia, both two and four-cycle engines are employed, and there is a tendency towards the use of engines of greater weight and more moderate speed. In England the two-cycle engine has not found favor. Practically all of the English boats are equipped with four-cycle engines of moderate speed and weight. No effort appears to have been made to gain a slight increase in surface speed by the use of the lightest possible engines. In the United States both two and four-cycle engines have been installed, and in a number of cases the two-cycle engines have been of the high-speed light-weight type. The tendency now is towards the use of a heavier engine, of more moderate speed, a development which means an increase in the displacement of the boats in order to avoid a loss of surface or submerged speed.

The lightest Diesel engines so far employed weigh about 50 pounds per brake horsepower and operate at a speed of about 500 revolutions per minute. The heaviest engines weigh about 100 pounds per horsepower and operate at a speed of about 350 revolutions per minute. A fair average for single-acting engines of substantial construction and moderate speed may be taken as 70 pounds per horsepower. While future development will no doubt result in some decrease in weight without sacrificing durability or reliability, there is no reason to believe that the improvement in single-acting engines will be radical. Internal-combustion engines in general, however, have not yet by any means reached the limit of their development, and it is not unreasonable to assume that it will be possible to produce units of this type of great power of substantially the same weight and space as that required for steam installations, and comparing favorably with the latter with respect to the matter of reliability.

In the present stage of development of the single-acting Diesel engine it is entirely practicable to install a submarine power plant of from 4,000 to 5,000 horsepower in a boat of from 900 to 1,100 tons surface displacement. When this displacement is coupled with a length of about 260 feet, a speed of from 20 to 21 knots can be obtained with the above power. When we come to consider the question of a speed of 25 knots or more, it is well to remember that excessive length in a submarine is in itself a very undesirable feature. In operating submerged it is not always possible to avoid some fore-and-aft inclination of the boat's axis; and when great lengths are involved comparatively small angles of inclination result in great differences in the level of the bow and stern. Obviously, then, the greater the length the greater the depth of water required for rapid and

safe submerged maneuvers. On this account, having in view the possible and probable uses of the fleet submarine, every effort will no doubt be made to keep the length within reasonable limits. Adopting, say, 300 feet as the limit of length, and coupling this with a displacement of about 1,200 tons on the surface, from 10,000 to 12,000 horsepower would be required for a surface speed of 25 knots. An installation of Diesel engines of such power would at this writing be experimental. It follows that, notwithstanding its serious disadvantages for submarine service, submarine designers may in the immediate future be forced to employ a steam plant in order to meet the demand for a surface speed of 25 knots or more. With steam-driven vessels, to secure equivalent radius of action, the fuel supply must be at least double as compared with Diesel-engined vessels. Taking this into account it would appear that a speed of 25 knots, coupled with a radius of, say, 1,500 miles at full speed and 3,000 miles at 14 knots, cannot be accomplished on the above limit of displacement without a material sacrifice in submerged qualities. To illustrate this let us take a concrete case. Starting with a Diesel-engined vessel of 1,100 tons surface displacement and 20 knots speed, coupled with a normal fuel radius of 1,500 miles at full speed and 3,000 miles at 14 knots, and allowing 70 pounds per horsepower for the main engines, we arrive at a weight of approximately 220 tons for main engines and fuel. Now considering a steam-driven 25-knot boat of the same radius of action and corresponding speed, and assuming that the machinery installation proper can be made on 45 pounds per horsepower, we arrive at a total weight of about 366 tons, or an increase of 146 tons. The increase of 100 tons in displacement will involve at least an increase of 40 tons in the weight of the hull and fittings, leaving us a net increase of not more than 50 tons. Hence, if we are to keep within the limits of the displacements mentioned, we must sacrifice the submerged qualities. Inferior submerged qualities are certainly serious defects from a military standpoint, but a danger of an even graver defect than a moderate decrease in submerged speed and radius is certainly involved in the installation of steam machinery. It has been pointed out above that rapid submergence is an absolutely essential quality, and in this respect steam-driven submarines must inevitably be inferior to those driven by internal-combustion engines. Whether and to what extent the inferiority of the steam plant in this respect can be overcome remains entirely a question for the future, but it is certainly a fact that at the present time the difficulties have not been overcome in France, even in installations of comparatively small power. For the above reasons the steam plant, if adopted at all, will be considered as a temporary expedient, and the steam engine is certain to be displaced by Diesel engines as soon as the development of the latter permits the attainment of the surface speed required. Because steam units have been developed for other purposes to powers in excess of those which the submarine requires for a speed of 25 knots, it is often assumed that there will be no great difficulty in adapting the steam engine to submarine propulsion. In the author's judgment this is an erroneous assumption, since, as a matter of fact, the adoption of the steam drive to the requirements of the submarine involves many a step in the dark, and the attempt is likely to result in many disappointments. In fact, as indicated above, it is not beyond the bounds of possibility that the effort may result in complete failure, in that while the surface speed may be obtained, other and vital military qualities may be sacrificed to such an extent as to make it impossible for the vessels to perform the service for which they are designed.

It had been the author's intention to supplement this discussion with some account of the efforts being made in Europe to develop the fleet submarine, and also to touch upon the possible effects upon such designs of the scouting and commerce-destroying duties which might well be performed by this single type. The author regrets that a lack of time requires

the termination of the paper at this point, and at the same time presents his apologies to the members of the society for the sketchy nature of the paper, which has been unavoidable under the circumstances of its hurried preparation. In view of the very great public interest in, and importance of, this subject at the present time, intelligent discussion by those capable of realizing the difficulty of the problems to be met is greatly to be desired, since from such discussion helpful suggestions are almost sure to flow. Should this brief survey of the present situation serve to provoke such discussion, the author will feel fully repaid for the work of preparation.—"Engineering."

THE SUBMARINE POWER PLANT.

By Allen Hoar.*

The first operable means of motive power installed in a submarine was the steam engine, but this was later abandoned, and it is considered that no practical solution of the problem of power was reached until the advent of the internal-combustion engine. It is true that the present-day development of the submarine boat started from the adoption of this means for propelling power. Strange as it may seem then, the tendency at the present time is to return to steam power for this purpose. There are good and sufficient reasons for this tendency, however, which will be discussed at some length later on.

The gasoline engine was the first of the internal-combustion engines to be adopted and these are installed in the A, B, C, D and three of the G class boats of our Navy. Owing to the high state of perfection and reliability that these engines have reached in the last few years they have given very satisfactory service on these boats, and could be expected to give even greater satisfaction at the present time, in view of their more recent development and improved efficiency. The gasoline engine afforded a relatively light compact form of prime mover and was structurally simple and easy of repair. Its ailments were easily understood and quickly remedied, and in fact, today it has become so generally well known that almost any young boy can manage one. The gas engine, however, had its drawbacks and these were: the cost of operation due to the high price of gasoline; danger of fire and explosion from gasoline vapors, and danger of asphyxiation by escaping carbon dioxide gases or poison by carbon-monoxide gases. The first of these objections is real, but the others, in so far as they can be easily remedied and eliminated, cannot rightly be considered so. In fact, the author knows of no serious accident primarily due to their cause. Gasoline, it must be admitted, is a great searcher, but still it should be possible to construct tanks sufficiently tight to hold it, and in any event, by means of a proper system of ventilation, the probabilities of any such mishaps are at once eliminated.

For those reasons, however, in all of the later boats the gasoline engine has been superseded by heavy-oil engines of the Diesel type. The main reason for this change, and there can be no other, is on account of the great economy of fuel consumption of the Diesel engine. This engine burns a cheap fuel, almost any low-grade oil which can be vaporized, and it will develop one B.H.P. per hour on from .55 to .63 of a pound of fuel. This is, of course, a great feature in favor of it. Some of the engine dealers guarantee a fuel consumption of as low as .48 of a pound of fuel per B.H.P. per hour, but in practice I do not know of any case where this economy has been attained.

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Theoretically the Diesel engine is extremely simple, but in actual practice and construction it is exceedingly far from being so. The theory of the Diesel principle is that on the first down-stroke of the piston the cylinder is filled with fresh air, which is then compressed by the following up-stroke of the piston to a pressure of about 500 pounds per square inch. The compression raises its temperature to about 1,000 degrees F. At the instant the piston reaches the top dead center a small quantity of fuel oil is injected along with a fresh quantity of air at a pressure of about 900 pounds per square inch. This takes place usually through a period of about one-tenth of the downward stroke of the piston. The fuel, which is broken up into a fine spray by the pressure of the air entering with it, immediately it comes in contact with the hot air within the cylinder, starts up combustion due to the temperature of the contained air being much higher than the flash point of the oil. This combustion continues through part of the stroke, supposedly until all of the oil has been completely burned, and expansion takes place during the remainder of the stroke on account of the expansive force of the pressure and temperature of the gas. In the first part of the stroke the aim is to have the combustion proceed at a rate which will cause the volume of gas to increase in the same ratio as the volume of the cylinder during combustion so as to keep the pressure constant until the fuel is completely consumed. At the end of the stroke the exhaust valves are opened and the burnt gases are pushed out by the following upward stroke.

The foregoing statements apply more particularly to the four-cycle Diesel, but the principle of the two-cycle is essentially the same; the chief difference between the two being that in the two-cycle engine the exhaust of the burnt gases and the fresh charge of pure air take place at practically the same time when the piston is near the bottom of the stroke. In some designs the scavenging air is admitted at the top of the cylinder through valves in the cylinder head and so blows the burnt gas down and out the exhaust ports at the bottom of the cylinders. These ports are uncovered by the piston as it nears the bottom dead center. Other designs do away with all valves but the air-starting and fuel-injection valves in the cylinder head. In this case the scavenging air enters through ports in the side of the cylinder, which are uncovered shortly after the exhaust ports situated on the opposite side of the cylinder. With the two-cycle, then, the valve gear is much simplified and a great deal of very exasperating valve trouble is done away with, but, on the other hand, the scavenging air for the two-cycle must be injected under pressure, usually, about nine pounds, which necessitates the addition of a low-pressure air compressor and greatly complicates the mechanism of the machine. The economy of the two-cycle is also much lower than in the four-cycle.

The primary cause for the serious difficulties which are to be met with in the Diesel engine, is the excessive temperatures that are generated in its cylinders,—the maximum temperature reached being about 3,000 degrees F. This high temperature, together with the high pressure in the cylinder, imposes two distinct conditions which must be met by the designer in calculating the stress upon the walls. These conditions apply also to the cylinder heads and pistons. It is quite conceivable that a thick cylinder exposed to high temperature and to a high inside pressure as well will be actually stressed more than a thinner one. It is known that cast iron appreciably changes its form and dimensions when submitted to continuous high temperatures, and develops a state of high internal compression as it tends to expand. This tension is, however, unevenly distributed throughout the metal on account of the cooler outer surface, thus accentuating the detrimental effects upon the metal. The answer to this difficulty seems to be one of metallurgy. A new metal must be developed, one having properties of greater tensile strength, little distortion from

excessive heat and still be no heavier than iron, before the Diesel engine can be successfully adopted for submarine use.

I do not mean to decry the Diesel engine as a type, for it has proven to be an ideal form of prime mover in stationary practice and has met with considerable success in ordinary marine practice, but in both of these cases the conditions encountered are as dissimilar to those encountered in a submarine as can be.

The ideal submarine engine must be a high-speed powerful engine of comparatively light construction, simple, accessible and reliable. The necessity of these qualities may be readily understood by referring to figure 1, which clearly shows the cramped and limited quarters in the interior of the machinery space of a submarine. The engines, in this case, gasoline, are shown forward of the motors and jambed back against the frames of the hull, where it is exceedingly difficult to make any repairs.

The proper solution of the Diesel engines for reliability and accessibility will permit neither high speed nor light weight to enter into their characteristics, unless its extremely high temperatures and pressures are first considerably lowered. It is quite possible that this may be effected to some extent, but there seems to be no concerted effort in this direction. Simplicity, however, is a quality which is highly improbable will ever be reached in this type of engine on account of its numerous troublesome auxiliaries.

In figure 2 is shown the complicated compressor end of a two-cycle engine. Figs. 2, 3 and 4 all show the complicated valve mechanisms and the cylinder-head castings.

In the E and the F class boats light four-cycle Diesel engines were installed, but have never given satisfaction. Primarily the cause of their trouble is the inadequacy of the construction. Attempting to keep the weight of these engines down to within certain limitations, they were constructed of built-up sections of plates and angles riveted together. The results of this manner of construction might well have been foreseen; they have simply shaken to pieces and set up new difficulties which serve to accentuate the inherent troubles of the Diesel principle. Both the H and K boats have had their share of engine trouble, but it is hoped that in the newer boats many of these difficulties will be overcome.

In view of all this, then, is not the present reliability of the gasoline engine to be greatly preferred to the economy of the Diesel engine? In time of war, the purpose for which these boats are constructed, it seems to me that efficiency is the required object to be attained no matter what the cost. If the gasoline engine be attended with other risks, are the dire possibilities of these risks any greater than the unreliability of the Diesel engine? A fair comparison of the two types of engines may be had by considering for the moment the D class of boats, some of the last to be fitted with the gas engine, and the performance of any of the later boats. The ever readiness and general efficiency of the D boats is to be favorably compared with any boat in our own or any foreign navy.

Much comment has been made of late upon the generally considered remarkable performance of the German submarines in the present war and of the apparently successful results shown by the Krupp Diesel engines installed in these boats. On the contrary, however, the Krupp engine is no more successful than any of the other submarine types of Diesels, and, in fact, is by far from being the best of these. Previous to the outbreak of the war the German submarines were kept pretty close to the repair shops and tuned up to their highest pitch ready to be let loose with the other dogs of war at instant notice. This fact alone accounts for their successes. In fact the German submarines work in relays, a certain number being on duty while the others are being overhauled at the bases. In this way, after each cruise a boat is given a

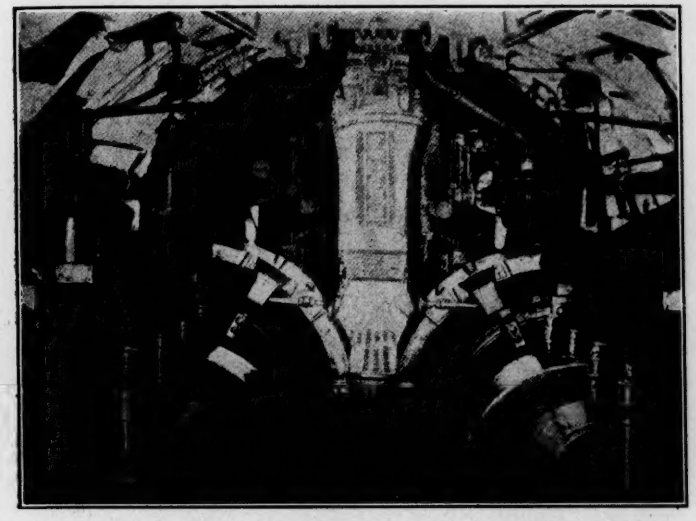

FIG. 1.—MOTOR ROOM OF A SUBMARINE (G-4), LOOKING FORWARD FROM THE AFTER PART OF THE VESSEL—IN THE FOREGROUND ARE THE TWO ELECTRIC MOTORS WHICH DRIVE THE SUBMARINE WHEN IT IS SUBMERGED. FORWARD OF THESE MAY BE SEEN THE MAIN ENGINES.

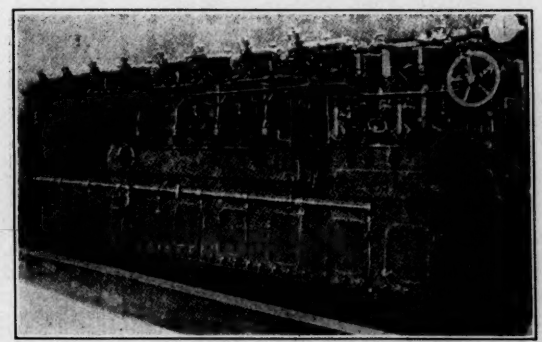

FIG. 2.—A 900 B.H.P. NURNBERG SUBMARINE MOTOR. SHOWING THE COMPLICATED COMPRESSOR END FOR THE INJECTION AIR. THIS IS A TEN-STAGE COMPRESSOR FOR DISCHARGING AIR AT 900 POUNDS.

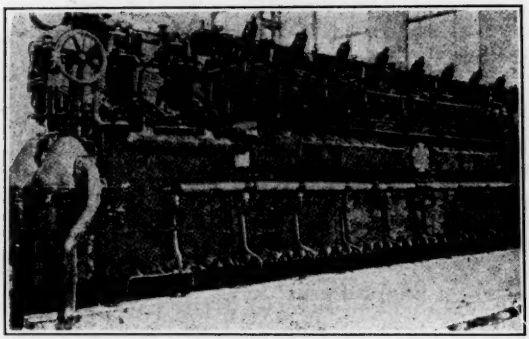

FIG. 3.—A TWO-CYCLE NURNBERG SUBMARINE MOTOR. 8-CYLINDER 900-B.H.P. NOTE COMPLICATIONS OF CYLINDER HEADS.

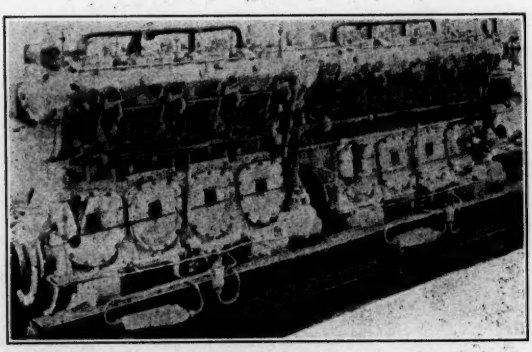

FIG. 4.—A THORNYCROFT SUBMARINE ENGINE. BUILT EIGHT YEARS AGO FOR THE ITALIAN NAVY. TWO 4-CYLINDER ENGINES ARRANGED TANDEM.

thorough overhauling and is therefore ready for work when her turn comes to put to sea again.

MOTORS AND STORAGE BATTERIES.

At the present time all submarines are propelled under water by electric induction motors, the electrical energy being supplied from accumulator cells. Big advancement has been made in the design of the electrical equipment for submarine installation, and especially in the method of control.

The present motors are ruggedly built, have their armatures mounted upon the main shafting of the engines and are well insulated. They are of the interpolar, direct-current, ventilated type, capable of running in either direction and under variable load without adjustment of the brushes. A potential difference of about 70 volts is allowed at the field terminals to provide for speed regulation when running as a motor and for adjustment of voltage when running as a generator. They are often run at an overload of as much as ninety per cent. without injurious heating.

The first controls to be used were plain knife switches. These are now all enclosed to eliminate the danger of sparking, and in some cases oil baths provided. The starters for the main motors are of the contactor type, master-drum control with interlocking features. This type is advantageous in that it permits the location of the control to be had in the most convenient place. Its only drawback is the complexity of its construction, but with the high voltage now handled it has become an absolute necessity. Automatic circuit breakers of the latest type are provided wherever necessary.

STORAGE BATTERIES.

Although the efficiency of the motor has been greatly advanced the problem of the storage battery still remains one of much dissatisfaction and it is quite improbable that the inherent defects of it will ever be overcome.

There are two distinct types of storage batteries in general use at the present time; the first is known as the lead battery and the second as the Edison battery.

The lead battery is the only type that has been used aboard submarines up to the present time. I understand, however, that there is now one, if not more of the boats, having the old batteries replaced by the Edison cells.

The lead batteries, as their names would imply, have active plates of lead material using sulphuric acid of a density of about 1.23 as an electrolyte.

There are several methods of manufacturing the lead plates, the three forms best known being the Planté plate, the Pasted plate and the Ironclad plate, this latter being a particular form of the Pasted plate.

The Planté is manufactured with the lead made into a fine grid which is cast, grooved, or spun in such a way as to afford a large superficial area for the electro-chemical action to take place upon. The grid is then subjected to this electro-chemical process which reduces the exposed lead surface to peroxide of lead for the positive plates and to spongy lead for the negative plates.

The Pasted plate is manufactured by pressing a pasty composition of lead and a small percentage of antimony into the annular spaces of a structural frame formed of network. The plates are then subjected to an electro-chemical process as before, reducing the plates to peroxide of lead for the positive plates and spongy lead for the negative plates.

The Ironclad plates are formed into positive plates only. They con-

sist of hard-rubber compound frames supporting hard-rubber tubes set side by side. The active material is formed by running antimony lead rods full length in the center of the rubber tubes, which are perforated, and otherwise filling in the tubes with red lead. It is then reduced into a positive plate by electro-chemically reducing the material to peroxide of lead.

In the batteries of the submarines the Planté positive plate is used in combination with the Pasted negative plate, or else the Ironclad positive plate is used with the Pasted negative plate; these combinations seeming to give the most satisfactory results.

The Edison storage battery uses nickel oxide as the active material for the positive plates and iron oxide for the active material for the negative plates. The electrolyte used is caustic soda.

The positive plates are made up of a steel frame supporting perforated steel tubes which contain a quantity of nickel hydrates for forming the active material.

The negative plates are made up of two perforated steel sheets forming pockets between them in which is contained iron oxide for the active material.

In addition to the enormous amount of weight, in round numbers about 59 tons, and the valuable space which it occupies, the lead battery is objectionable upon the score of its inherent dangers. There is the ever present danger of explosive gases collecting with the contingent result of battery fires and terrific explosions, the only means of fighting which seems to be to leave the ship and let them burn. There is also the continual danger from the generation of chlorine gas, which is a deadly poison and which is liable to be generated at any time if salt water finds its way to the batteries, and, lastly, the danger to the hull itself from leaking or the slopping over of sulphuric acid which immediately attacks the steel plates of the horseshoe tank, and, unless the installation has been made in such a way as to afford perfect inspection, which is not the general case, the metal is soon eaten through by the chemical action of the acid.

The advocates for the Edison battery are claiming for this type the entire elimination of all these bad features of the lead battery. This, however, is not true, for the Edison battery is quite as liable of generating the explosive gases and having fires as is the lead battery, although it is free from the deadly fumes of chlorine gas and the trouble of the leaking acid.

On the other hand, the lead battery has an average discharge voltage at the three-hour rate of about 1.83 volts per cell, whereas the Edison battery has at the three-hour rate of discharge of but from 1.1 to 1.2 volts. This, then, would mean that with the Edison battery the number of cells would have to be increased about 60 per cent. and would require considerably more floor space.

The weight of the Edison battery is also much higher than that of the lead battery, and this is an all important factor. In view of this fact, then, and that the Edison battery is less than 72 per cent. as efficient as the lead battery, it would seem that to install new equipment that requires more weight and space than that which is already installed, and which therefore must necessarily detract from the efficiency of other factors now contained, would be far removed from the ideals that we are trying to gain in submarine development, because it would be in this case making a sacrifice of other factors and only getting for our trouble a decreased radius of action.

The cost of the Edison battery is much more than the lead battery, but on this score the life of the Edison battery greatly exceeds that of the other, so the price may be conceded to be in favor of the Edison battery, if anything.

The present reversion to the steam engine as a means for surface propulsion is brought about by the inherent difficulties found in the heavy-oil engine, and because now the steam engine has reached a state of efficiency and reliability found in no other form of prime mover.

And by the development in size of the submarine boat, which, of course, demands a correspondingly increased horsepower, it is found to be impractical, in its present stage of development, to install Diesel-engine sets on account of the limited space and weight available to the power plant.

As far as economy is concerned, by combining the use of high-pressure steam with a high degree of superheat and using a high mean referred pressure it is quite possible, by using an improved form of oil-burning boiler to secure an economy as low as .7 to .8 of a pound of fuel per B.H.P. per hour. This is but little in excess of the Diesel-engine consumption, and by using the steam engine the attendant improvement in conditions would be manifold. The weight for the steam engine for the same power of Diesel engine is much less. The mean effective pressure is higher in the steam unit and its cylinder is subjected to but little more than one-fourth of the extreme pressure and to not one-fourth of the extreme temperature of the Diesel unit. The steam unit is also the simpler and more easily accessible of the two.

The ideal form of power plant for the submarine, as may be easily understood, is one that is capable of operation both when on the surface and when in the submerged condition, that is, in other words, a single unit which will do away with the present dual system. The problem, then, is to find some method by which the prime mover to be used for surface propulsion can be made to furnish the power for submerged work, thus doing away with the present storage-battery system and its inherent dangers and limitations.

The problem has been attacked by many in the last few years, notably among them, an Italian engineer by the name of Del Proposto, and a Spanish engineer named D'Equevilley.

The Del Proposto proposition is essentially an air proposition, using the internal-combustion engine to propel the boat and to drive an air compressor for storing up air in tanks when on the surface. In the submerged condition the mechanical energy of the stored air is used back through the compressors and through all or part of the cylinders of the internal-combustion as air motors for the propelling power. It is understood that Del Proposto built a boat and had his system installed. But little is known of the results obtained, however, and it is believed that the performance of the equipment did not come up to the expectations. His troubles would evidently be mechanical difficulties resulting in inefficiency.

The D'Equevilley proposition is that of a soda boiler, using the steam generated from a process of slaking caustic soda. When the vessel is about to submerge the exhaust steam from the engines is turned into this soda boiler, producing a secondary steam caused by the action of the soda in absorbing the water vapor. The heat evolved by this action forms a secondary steam which is used in the engines and the cycle continues. This process goes on until the caustic soda has become saturated, when the vessel returns to the surface and the soda is reconcentrated.

A boat of this type has been built in France and the process has also been tried out in Germany, but it is not believed that any great success was obtained in the trials. The principle of this system is old, having first been tried out in this country by Prof. J. H. L. Luck, of San Francisco, in his submarine the *Peacemaker*, in, I believe, 1873. The boat was built by the Submarine Motor Co., of New York. The same principle has since been laid before the Naval Department by a Chilian inventor, and it has also been worked upon by several engineers in Europe.

A system of propulsion has lately been perfected under patents held

by the L. A. Submarine Boat Co., of California. This system, with which the writer has been intimately connected and the development of which has been due to his efforts, provides means for utilizing the same power unit, both when on the surface and when running submerged.

Public policy will not permit of a detailed description of this system, but suffice it to say, that any form of prime mover may be used with satisfactory results. A boat has been built and put through trials under inspection by a Naval Board and the results have been found satisfactory in every way. By means of this equipment both greatly increased submerged speed and increased radius of action submerged are possible. The military advantages of these two factors are at once apparent.

At present it is rather much of a question to say what new turn the power plant for the submarine will take. It is highly probable, though, that if the Diesel engine is retained it will be of the non-reversible type, of heavier construction and will drive generators for supplying current to induction motors connected to the propeller shafts. This would permit of greater speed range and the elimination of starting and reversing troubles, as well as to cause the constructor less worry about the proper distribution of weights as effecting the trim of the boat. The size and weight of the motors could be kept down by using polyphase alternating currents of high frequency, or direct currents of high voltage, say 500 volts for surface work, and, if used with the storage batteries, a voltage of 250 for submerged work, as at present.

Or, perhaps, if we revert to steam we will find the solution in the turbine or in the turbo-generator sets. These would surely materially decrease the weight of the sets on account of greater rotary speed, and most certainly would add to the reliability of the unit as a whole.—"The Sibley Journal of Engineering."

SUBMARINE PROPULSION.

By Paul H. Berggeen, M. E.

The French engineer, d'Equevilley, has devised a type of machinery for submarine propulsion that can be used both on the surface and submerged, and his system is now being tried out in the French submarine *Charles Brun* and, according to "Applied Science," also in a German boat.

The apparatus consists of an ordinary boiler with oil fuel and a steam engine on the surface. When the boat submerges the exhaust steam is led to a concentrated lye of sodic hydrate which absorbs the steam under strong evolution of heat, thus serving as fuel in a secondary "soda boiler." This process goes on until the lye is saturated.

When the boat comes to the surface, and steam again can be developed in the primary boiler, the soda lye may again be concentrated by evaporation of the water which it has absorbed, and the boat is ready for the next submersion.

The plan offers the advantages that there is no change of propelling motor, the same engine being used under water as on the surface, and there are no products of combustion. The machinery can be forced without difficulty and relatively high power attained in either case.

While no electric motor is required, the system will, on the other hand, necessitate the addition of special soda boilers and a hot-water reservoir. The plant occupies so much space that the available weight can not be fully utilized; the center of gravity of the machine is high and requires extra ballast to be carried; the radius of action on the surface is naturally smaller than with an explosion motor, and there is likely to be a strong

corrosion of the boiler due to the effect of the soda. Nevertheless, the soda-boiler principle seems superior to any other existing power-plant project for submarines so far proposed.

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For underwater propulsion, electric power derived from a storage battery of lead accumulators still offers the best practical solution. Since the first appearance of these cells they have been improved upon in many technical details and are now reliable and durable. They will stand complete charging and discharging more than 400 times and may be expected to last about five or six years under ordinary service conditions. They allow great variations in power and are at their best at low rates of discharge, a most valuable means of submerging. They add to the stability of the boat as they can be stowed low, and they have a smaller space requirement than any other known apparatus for similar purpose, about four cubic feet per H.P. hour.

The American government is at present experimenting with accumulators of a different type, namely, the Edison alcalic iron-nickel cells, which seem to be superior to the lead cells in regard to durability, but more costly. The Edison cells consist of two sets of plates known respectively as "positive" and "negative." The plates of each set are connected together into one group. The positive plate is made of perforated nickel-plated steel tubes containing alternate layers of nickel-hydrate and metallic nickel-flake to secure perfect contact between the active material and the tube. The negative grid contains pockets of perforated steel holding the negative acting material, iron oxide. The electrolyte is a potash solution in water.

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While some French submersibles have been equipped with steam turbines for propulsion on the surface, the development of the submarine in its present form would have been almost an impossibility had it not been for the great progress in the construction of internal-combustion engines in general and the Diesel oil engine in particular.

The building of Diesel submarines started simultaneously in England, France and Germany, about eight years ago. The two-cycle engine is now invariably adopted by German builders, while the English are preferring the four-cycle engine, and the French are using both systems.

The Nürnberg engine is the most promising German make, being an eight-cylinder, two-cycle engine with two two-stage air compressors. It is designed for an output of about 900 B.H.P. at 450 r.p.m., and is original in having the scavenging pumps arranged as stepped pistons immediately below the working cylinders, so that there are in all eight such pumps. The piston-cooling arrangement in the engine is somewhat unusual, being carried out by pumping oil at a pressure of about 35 pounds per square inch through the crank-shaft, connecting rod and gudgeon pin into the piston. From here it returns to the oil cooler and is used over again. There is only one scavenging valve to each cylinder, and this is inclined to the axis of the cylinder, as is also the fuel-inlet valve on the opposite side. The method of reversing adopted in this case is to move the cam shaft through an angle of 30 degrees, which sets the scavenge valve and fuel valve at the correct positions for running in the opposite direction.

The design of the Fiat engine in Italy is similar to this type and adopted in recently constructed submarines.

The Sulzer submarine engine is perhaps the most novel of any type constructed. It is being built for the American and Japanese navies as a two-cycle, single-acting, six-cylinder machine, developing 600 B.H.P. at 400 r.p.m. The scavenging is here carried out, not by means of valves

in the cylinder head, but through ports at the bottom of the working cylinders. There are two sets of these ports, the upper one being controlled by means of a mechanically-operated valve, actuated from the crank shaft by means of levers. This allows extra scavenging air to reach the working cylinder after the main supply to the bottom port has cleared out most of the exhaust gases.

The Polar submarine engine, built in Sweden, is a four-cycle, six-cylinder engine with one two-stage air compressor. Its chief novelty in the construction lies in the arrangement adopted for rendering the motor directly reversible. In order to simplify this to the greatest degree it has been designed so that, in reversing, the exhaust valve and the inlet valve are changed over, the exhaust becoming the inlet and the inlet the exhaust. This arrangement is extremely simple for a four-cycle engine.

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The propulsion of submarines on the surface and in submersion by means of the Diesel engine and the electric storage batteries, respectively, while individually developed to a high state of perfection, does not solve the question of constructing a type of machinery that can be used both on the surface and submerged, especially where the propulsion under water should not entail any extra weight. Any process based on combustion involves the storage of atmospheric air or oxygen, but a storage of these gases in sufficient quantities will require excessive weight and space. Furthermore, the discharge of the products of combustion is liable to reveal the presence of the submarine. The above mentioned French soda-boiler installation will soon be able to show results which, if successful, will open a new era in the building of submarine engines.—"The Sibley Journal of Engineering."

APPLICATIONS OF GEARING FOR TRANSMITTING POWER,

WITH SPECIAL REFERENCE TO SELF-CONTAINED UNITS FOR REDUCING AND INCREASING SPEEDS.

By a Staff Correspondent.

During the past few years a large field has been opened to engineers whose business it is to deal with the application of gears for reducing and increasing speeds from modern high-speed and low-speed prime movers, in competition with belts, ropes, chains, etc., which in past years have been more generally adopted. Whilst these latter methods still have suitable application, the present-day perfection of machine-cut gears, with theoretically correct tooth profiles, has made the gear unit the only reasonable one to apply where considerations of space, efficiency and cost of up-keep are of any importance. With the increasing use of high-speed motors, the field of application for self-contained gear units has greatly enlarged, and it will no doubt be of great interest to briefly describe some examples of their application.

A great deal of highly technical matter could be written on this subject, but much can be gathered from a short article and descriptions of a number of photographs of actual installations, and it is intended to confine this article mainly to the consideration of self-contained units as power transmitters, and to leave for the present the more general application of gears to machines. A self-contained unit obviously means a unit in which all the stresses set up by the gears are taken care of by the unit itself, and which can be coupled direct to the motor at one end and the machine to be driven at the other.

Present-day users demand efficiency in operation and economy in service,

and as it is necessary to obtain these in the highest degree, and that the gear be protected from dust and foreign bodies and properly lubricated, and also that the shafts should be run in substantial bearings, properly and rigidly tied together, it will be readily apparent that the self-contained unit offers considerable advantages over gears mounted directly on to the shafts of the motors and the driven machines.

Fig. 1 shows examples of worm drives and worm and spur-gear drives combined, which are complete in themselves, and ready for connection to the machines to be driven, by any of the usual methods. By the use of such units the work of setting up and aligning the machines is very much simplified, and possible distortion of the various parts after the gears are put to work is avoided. Many of the old belt and rope drives from steam and gas engines are being replaced by high-speed motors and gear drives. Whilst the units shown on this photograph do not constitute the most efficient form of reduction gear, there are cases where the right-angle drives obtained by the use of worm gear is an advantage. Worm gear, however, is not as efficient as the more direct spur-gear drives.

It is often more economical to have a central generating or power house, and the necessary power required brought to each machine separately. The intervention of a gear reducing unit in many cases enables one to use a motor at its most economical speed, and as the efficiency of a spur reducing gear is extremely high, the combined efficiency of the motor and gear unit is higher than that of a specially designed slow-speed motor.

The simplest and cheapest form of reducing-gear unit is no doubt spur gearing, in which the two shafts are parallel, the gears being mounted on these shafts, which are run in suitable bearings. The forms of bearings which have been found most satsfactory for these gear units are, for fairly high speeds, ring-oiling bearings, fitted with cast-iron bushes lined with anti-friction metal. For slower speeds grease-lubricated bearings are more suitable. Some makers use for moderately high speeds ball or roller bearings. These may be satisfactory where the speeds are high and loads very low, but they are very liable to break down where shocks are encountered. Generally speaking, a single reduction straight-tooth gear cannot exceed 8 or 10 : 1 ratio, as this is about the greatest reduction which can be made economically; if higher ratios are attempted the pinions become extremely small, and the wheels very large, necessitating very large casings. The engagement of the teeth also becomes unsatisfactory, impairing the smooth running of the gear. For single reductions from electric motors it is a fairly common practice to mount the pinion direct on to the motor shaft, and in many cases this gears into a wheel mounted on an overhanging shaft. This method is liable to be extremely unsatisfactory unless the shafts and bearings are of very rigid design. If these points are overlooked when designing such a gear, although the gear may run satisfactorily at first, it will very soon become noisy and rapidly wear out.

If the best results are desired, all gears should be mounted with bearings as close up as possible to each side of both wheel and pinion. For higher ratios than 8 or 10 : 1 with straight-tooth gears it is necessary to use a double reduction. This type of case can be made of similar appearance to single reduction, and can have the high-speed and low-speed shafts in line, which, in a good many cases, is a distinct advantage. In both single and double reduction gears the casing forms the oil bath into which the large wheels dip, and lubricate the gears by splash; in both types rigid design is necessary to obtain the best results.

Double helical or herring-bone gears, as they are sometimes called, are completely ousting straight-cut gears where efficiency, long life and smooth running are of any consideration. They have distinct advantages over straight-cut gears, inasmuch as, size for size, they will transmit a

greater power, and consequently units with double helical gears are smaller than units containing straight-cut gears. They are much more silent in operation owing to the greater continuity of engagement of the teeth, and also a much greater reduction can be obtained with a single pair of geared as a less number of teeth can be used in the pinion, making it extremely small.

Fig. 2 shows a single reduction-gear unit with double helical teeth of the Chevron type, and with the shaft mounted in ball bearings, which have had to be very liberally designed. It will be noticed that the pinion is cut with shrouded teeth, and both wheel and pinion are cut by means of end mills. This method of cutting is expensive both in time and cutters, owing to the small diameters of cutters which must necessarily be used. This type also is more liable to errors in thickness of teeth than gears cut by one of the generating processes.

Fig. 3 shows a single reduction gear mounted on girders in actual operation in a machine shop. This particular gear is so arranged that either the right or left-hand line shafts can be driven independently or both can be driven together, as desired. This is obtained by means of the claw clutches clearly shown on the photograph. It will be seen that such an arrangement economizes space, and as it can be coupled direct to the shafts overhead, no floor space whatever is taken up.

Fig. 4 shows a similar type of gear to Fig. 2, mounted on a cast-iron bedplate, and coupled to an electric motor. In this case the bearings are of gunmetal, and are lubricated by means of chains. In this case, also, oil gages are shown indicating the height of oil in the casing, and also in the bearings. Whilst the method shown of lubricating the bearings is quite satisfactory for all ordinary cases, where the speed is very high and the pressures great, it is necessary to have some forced-lubricating system both for bearings and gears. A very important point to watch when erecting these gear units is to see that both the high-speed and the low-speed shafts are correctly aligned with the motor and the shaft of the machine to be driven.

Flexible couplings should in all cases be used if possible, but they should not be used with the idea of correcting inaccurate alignment. As thrusts from the shafts could be transmitted to the gears if solid couplings are used, flexible couplings will protect them from such outside thrusts. These thrusts may be set up in a long line of shafting through the action of heavy belts, or through a pair of straight-tooth bevel wheels on some part of the shaft tending to displace the line shaft endways, and if this thrust is transmitted through the gears, rapid wear may be looked for with consequent loss, inefficiency and considerably increased running cost.

Fig. 5 shows a pair of gears suitable for use in a casing, the pinion in this case being keyed to its shaft, and the shaft being reduced on the bearing portion.

Fig. 6 shows a pair of double helical gears with Chevron type of tooth, and it may be of interest here to point out that where double helicals possess a superiority over straight-tooth gears is in the number of teeth in engagement at one time. With straight teeth, and a small number of teeth in the pinion, only one tooth can be in engagement with the wheel at a certain time, and this tooth is in engagement along its entire length. With double helical teeth, more than one tooth is in engagement at any time, consequently the strength of a double helical gear is considerably greater than a straight-tooth gear of the same dimensions, and if a line be drawn across the face of the gear in the manner shown it will easily be understood that, whilst the top of one tooth is coming into gear, the next tooth will be further in gear, and the next tooth perhaps on the pitch line, and so on. From this it will be seen that the engagement of the teeth is continuous, whilst with a straight tooth, and a very small number of teeth in the pinion, there may be a distinct break between one

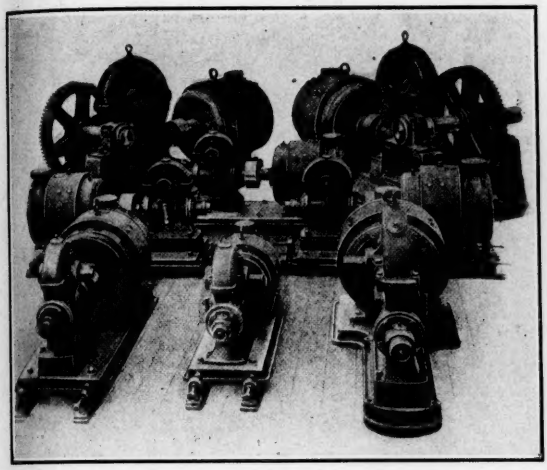

FIG. 1.—WORM DRIVES AND WORM AND SPUR GEAR DRIVES COMBINED. MADE BY THE POWER PLANT CO., LTD., WEST DRAYTON.

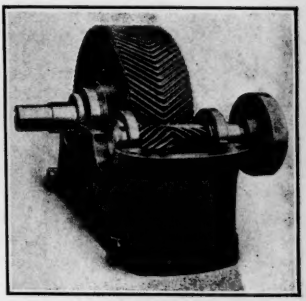

FIG. 2.—SINGLE REDUCTION GEAR UNITS WITH DOUBLE HELICAL TEETH. MADE BY D. BROWN & SONS, LTD., HUDDERSFIELD.

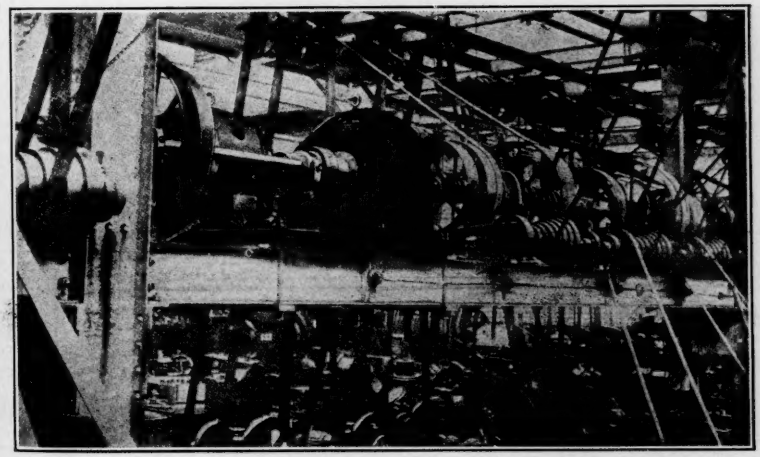

FIG. 3.—SINGLE REDUCTION GEAR MOUNTED ON GIRDERS. DAVID BROWN & SONS, LTD., HUDDERSFIELD.

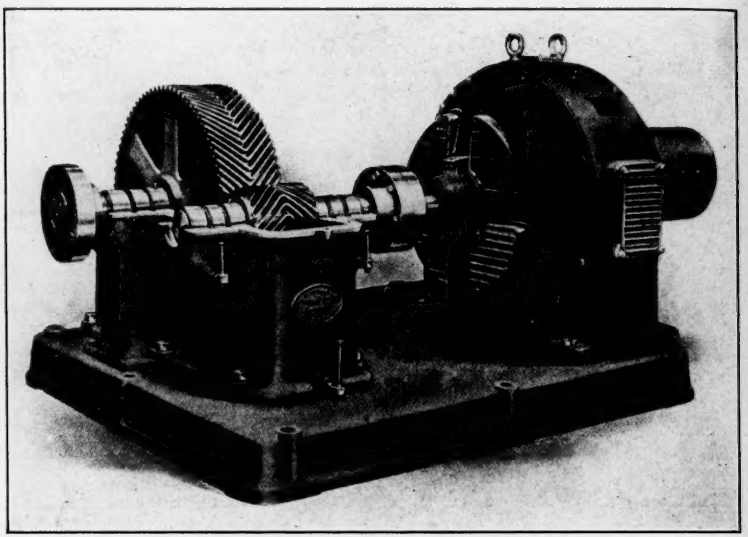

FIG. 4.—SINGLE REDUCTION GEAR COUPLED TO AN ELECTRIC MOTOR.

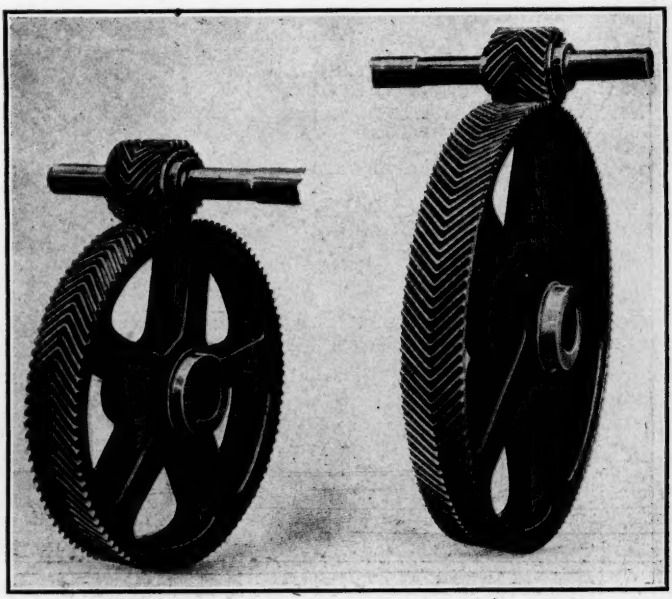

FIG. 5.—PAIR OF GEARS FOR USE IN A CASING. DAVID BROWN & SONS, LTD., HUDDERSFIELD.

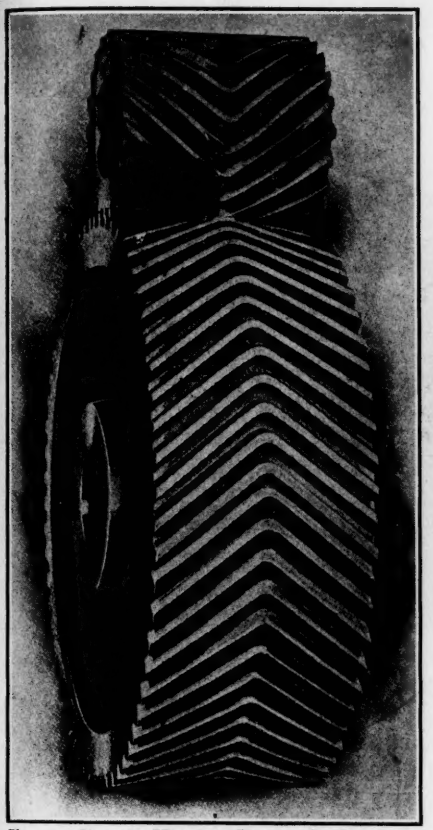

FIG. 6.—DOUBLE HELICAL GEARS. D. BROWN & SONS, LTD.

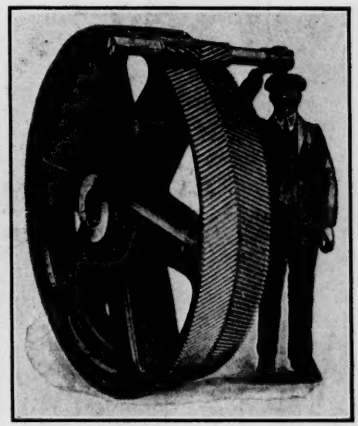

FIG. 7.—HIGH RATIO HAULAGE GEAR.

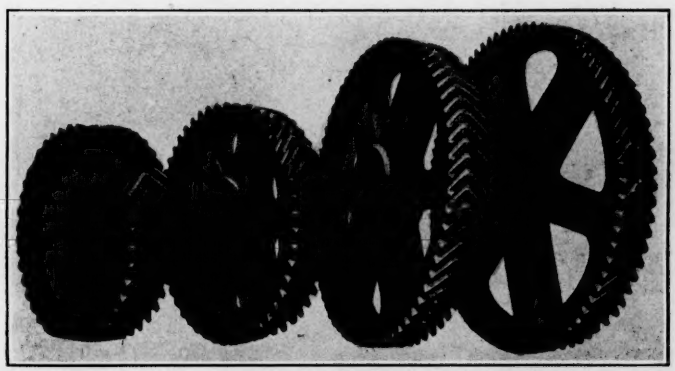

FIG. 8.—TEETH CUT BY THE END MILLING PROCESS. THE POWER PLANT CO., LTD.

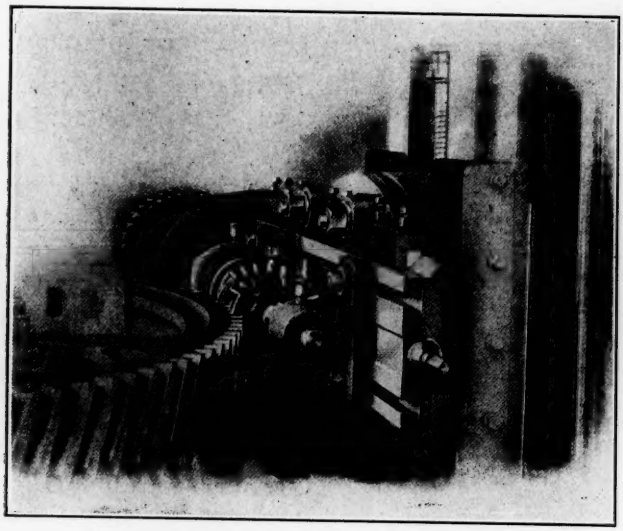

FIG. 9.—GEARS BEING CUT BY THE HOBBING PROCESS.

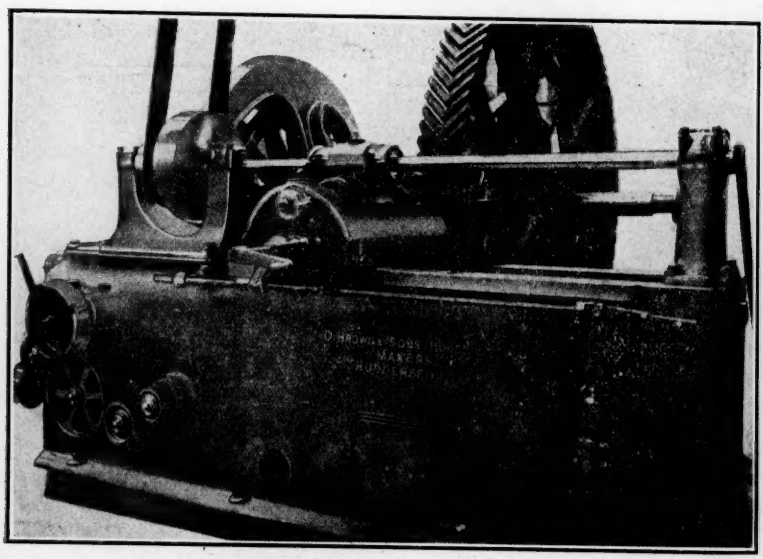

FIG. 10.—GEARS BEING CUT BY THE PLANING PROCESS.

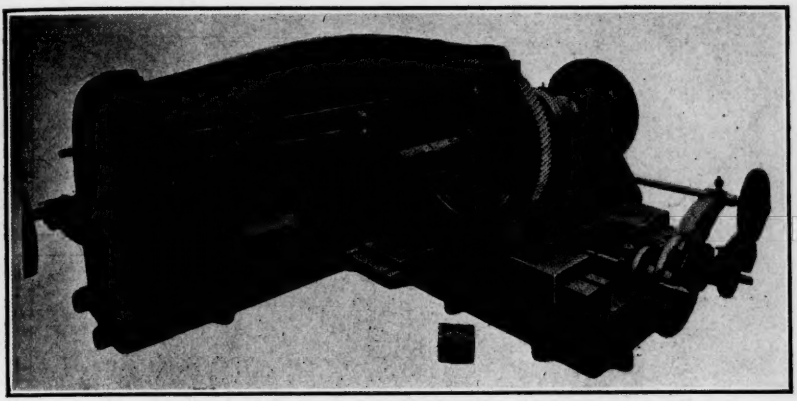

FIG. 11.—CUTTING GEARS BY THE END MILLING PROCESS.

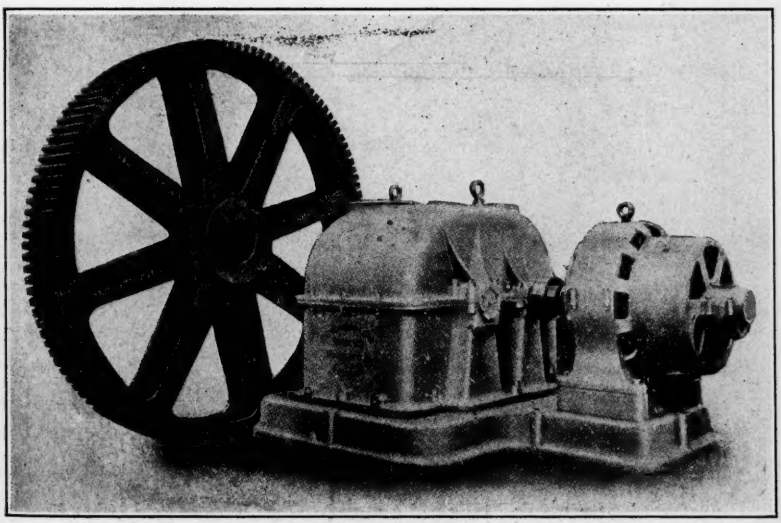

FIG. 12.—LARGE REDUCING-GEAR UNIT FOR A ROLLING MILL.

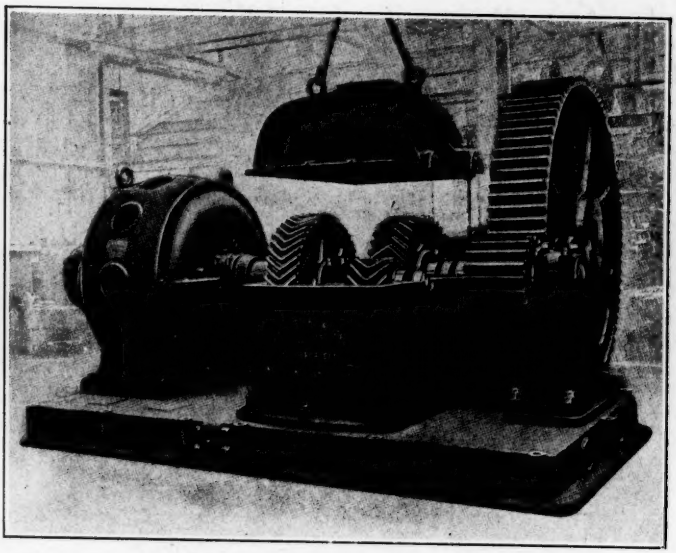

FIG. 13.—ANOTHER LARGE REDUCING GEAR.

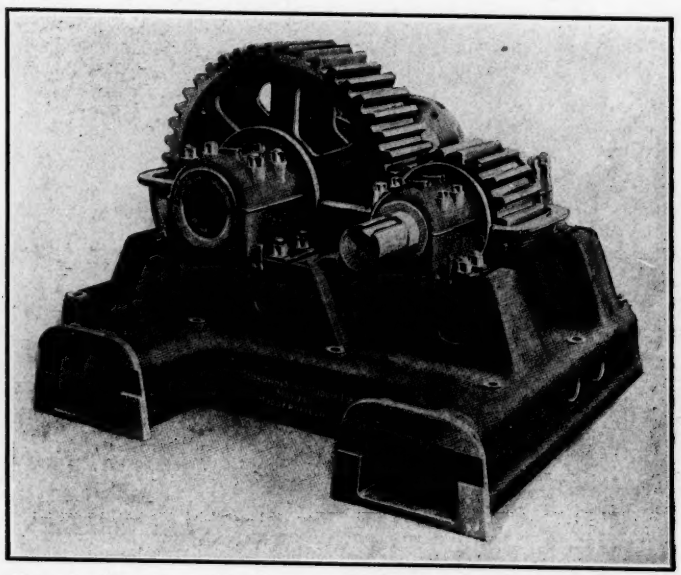

FIG. 14.—SPUR-GEAR UNIT FOR AN ELECTRICALLY-DRIVEN ROLLING MILL.

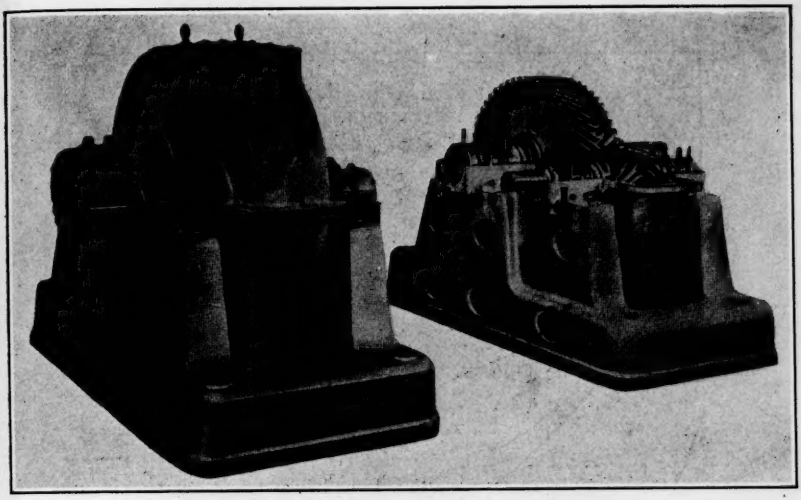

FIG. 15.—TWO DOUBLE HELICAL-GEAR UNITS.

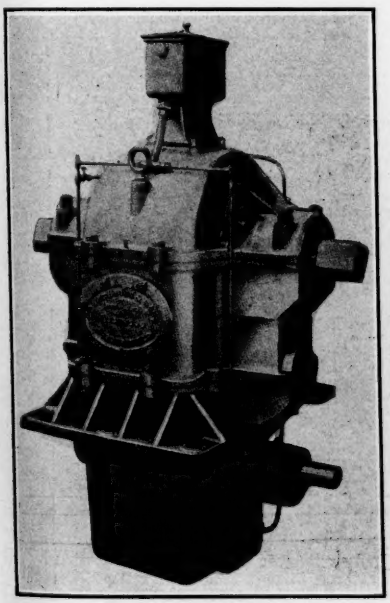

FIG. 16.—DOUBLE REDUCTION GEAR UNIT.

FIG. 17.—SINGLE REDUCTION GEAR.

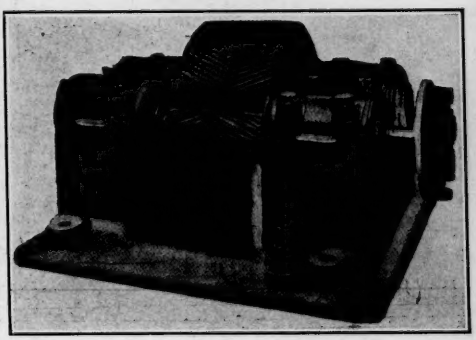

FIG. 18.—SMALL HIGH-SPEED INCREASING GEAR.
THE POWER PLANT CO., LTD.

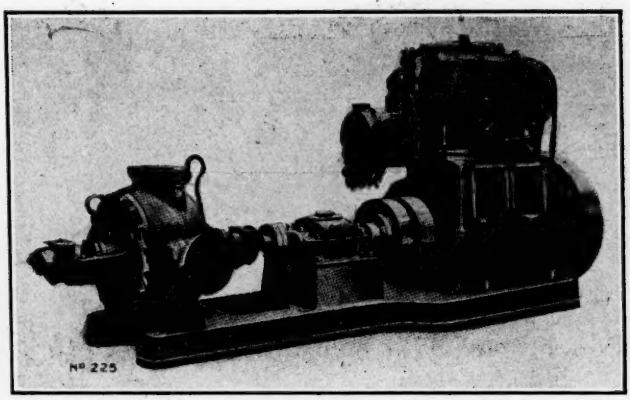

FIG. 19.—ANOTHER TYPE OF INCREASING GEAR CONNECTING AN OIL ENGINE TO A CENTRIFUGAL PUMP.

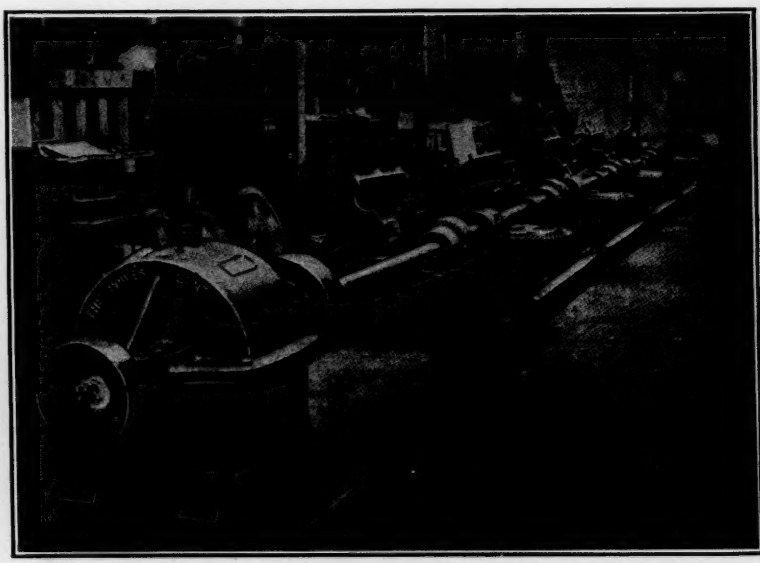

FIG. 20.—TOTALLY ENCLOSED DOUBLE REDUCTION-GEAR UNIT WITH HIGH AND LOW-SPEED SHAFTS. THE POWER PLANT CO., LTD.

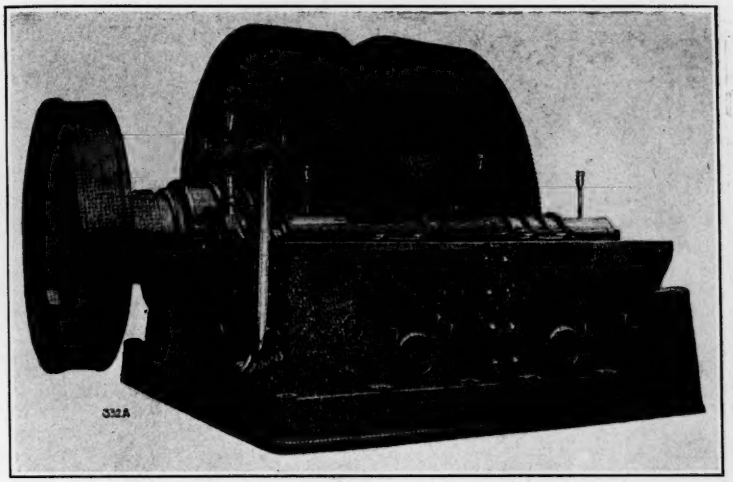

FIG. 21.—TURBINE GEAR UNIT. THE POWER PLANT CO., LTD.

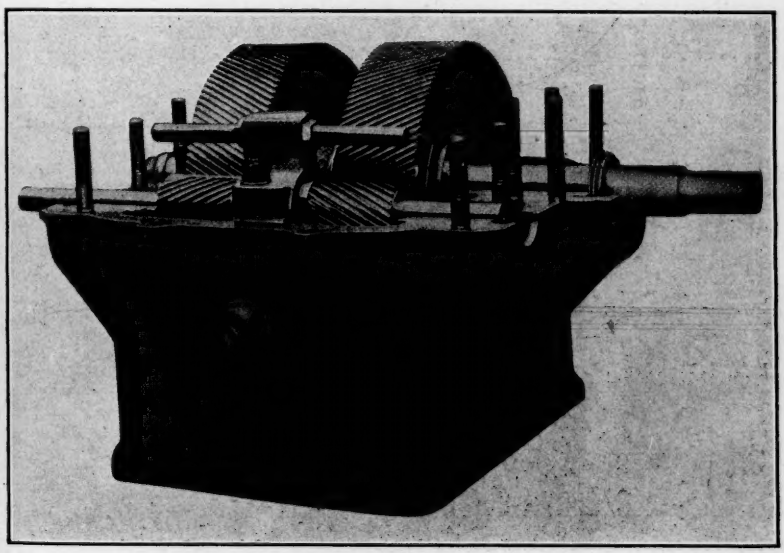

FIG. 22.—REVERSIBLE TURBINE GEAR WITH PINION ROTATING AT 13,000 REVOLUTIONS PER MINUTE.

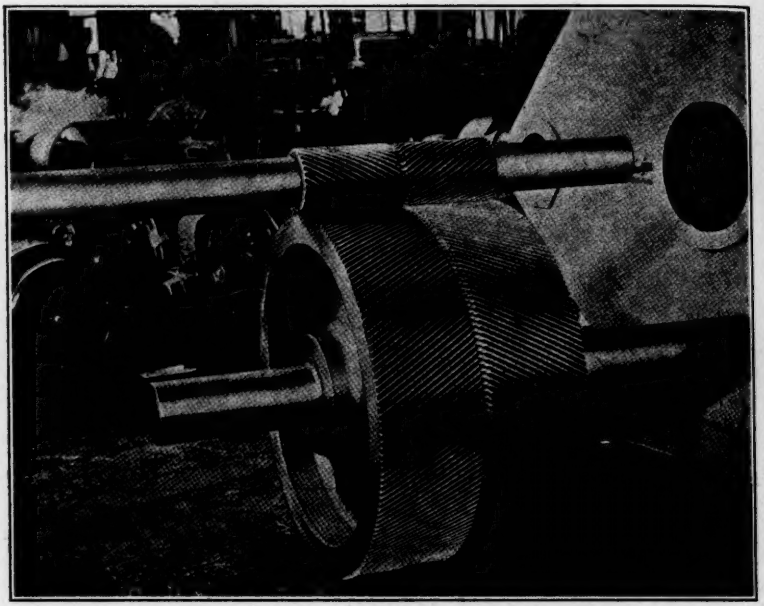

FIG. 23.—TURBINE GEARS. DESIGNED TO TRANSMIT 750 H.P. REDUCING FROM 3,600 TO 750 R.P.M.

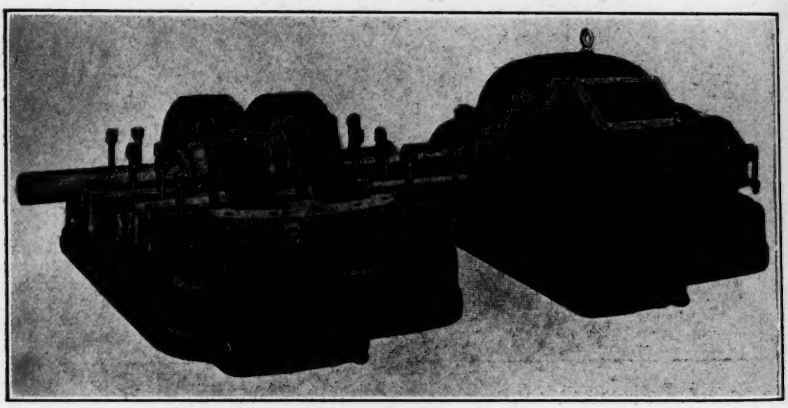

FIG. 24.—ANOTHER EXAMPLE OF TURBINE GEARS FOR DRIVING A CENTRIFUGAL PUMP FROM A TURBINE.

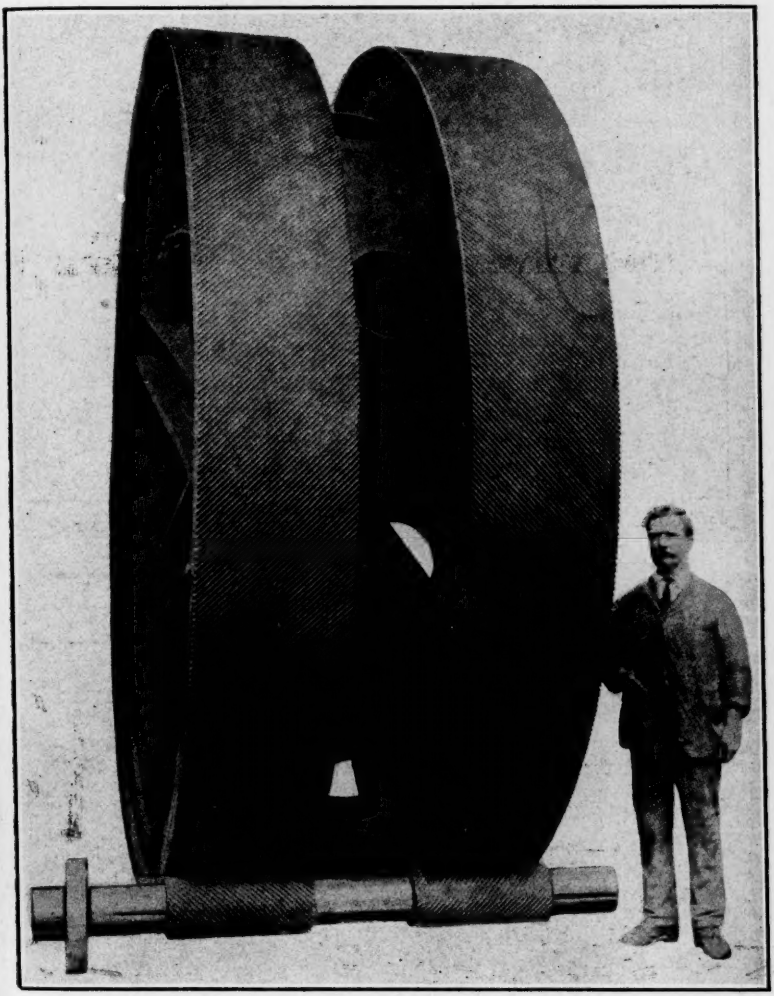

FIG. 25.—LARGE TURBINE GEAR FOR SHIP PROPULSION. MADE BY THE POWER PLANT CO., LTD.

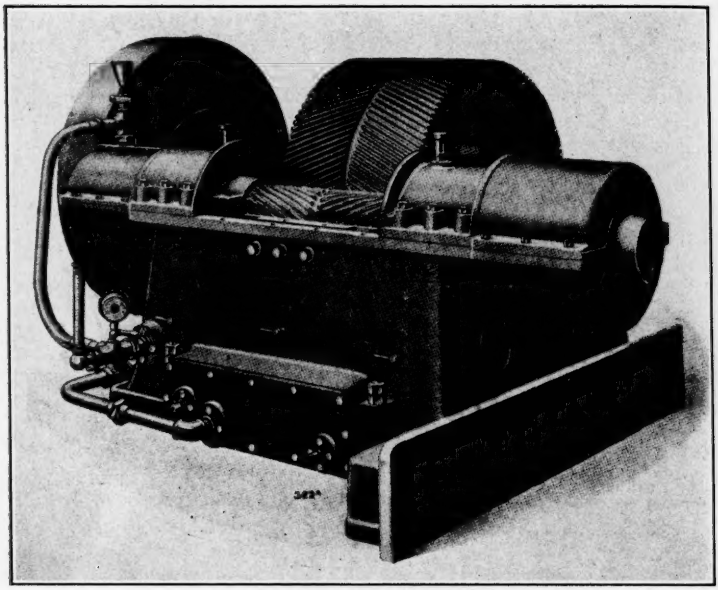

FIG. 26.—ANOTHER TURBINE GEAR WITH FORCED-LUBRICATION ARRANGEMENTS. THE POWER PLANT CO., LTD.

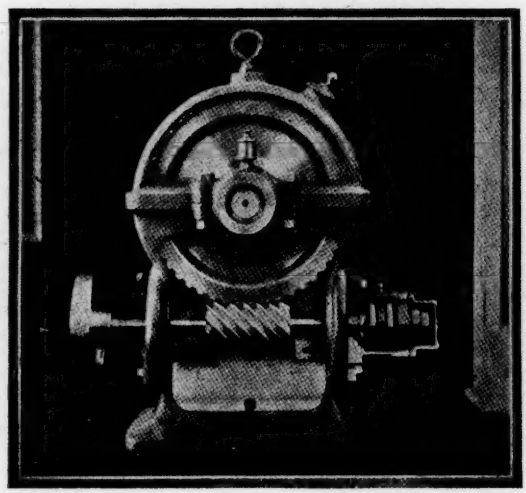

FIG. 27.—WORM GEAR UNIT. DAVID BROWN & SONS, LTD.

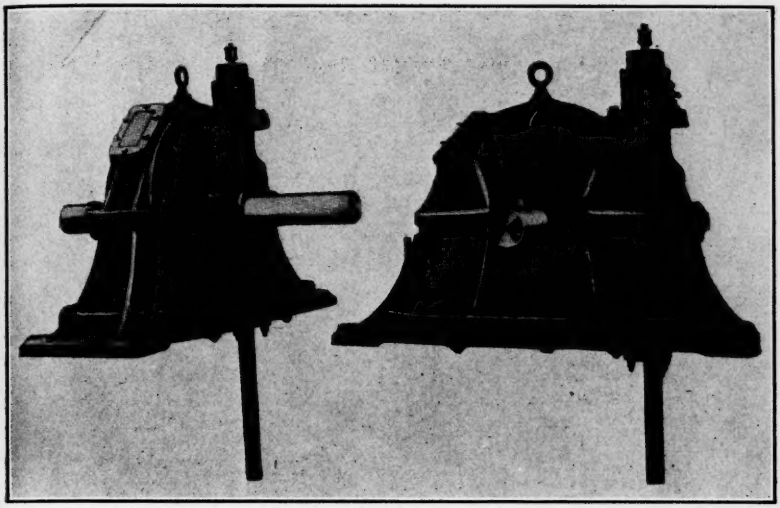

FIG. 28.—WORM-GEAR UNIT WITH VERTICAL WORM SHAFT AND HORIZONTAL WHEEL SHAFT.

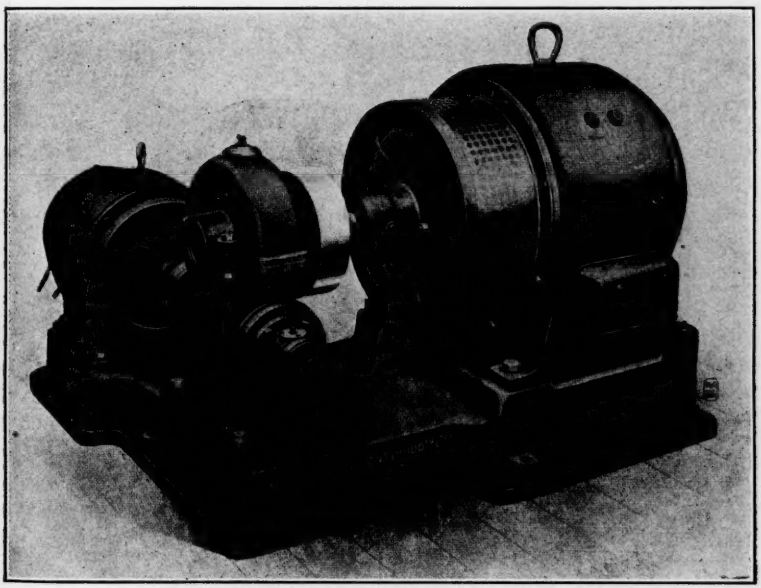

FIG. 29.—WORM-GEAR UNIT FOR STARTING A LARGE MOTOR.

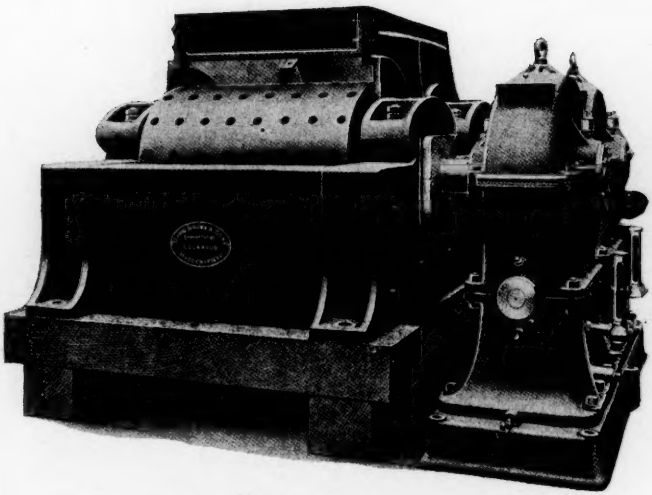

FIG. 30.—TWO WORM GEARS FOR DRIVING A SHALE BREAKER.

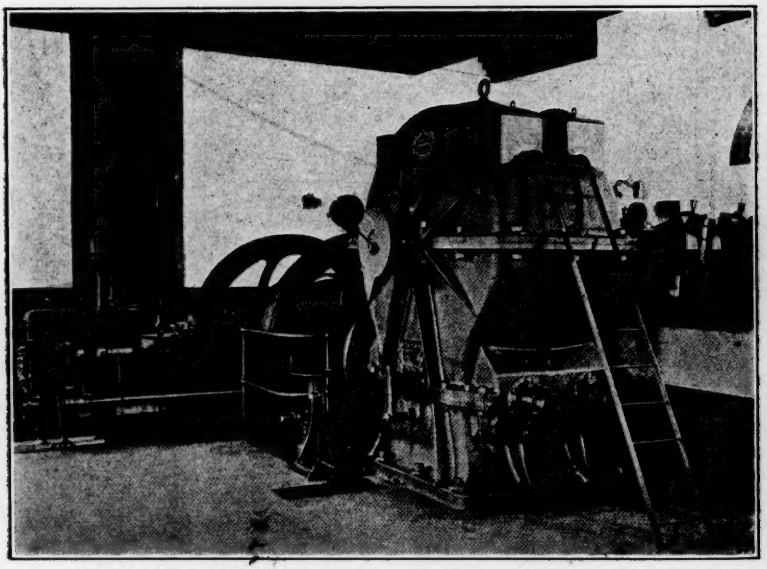

FIG. 31.—LARGE DOUBLE WORM GEAR. MADE BY DAVID BROWN & SONS, LTD., HUDDERSFIELD.

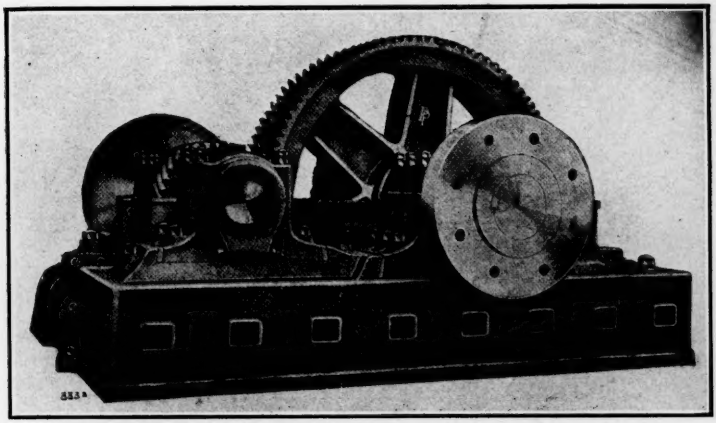

FIG. 32.—LARGE GEAR UNIT FOR DRIVING A BAR-ROLLING MILL.

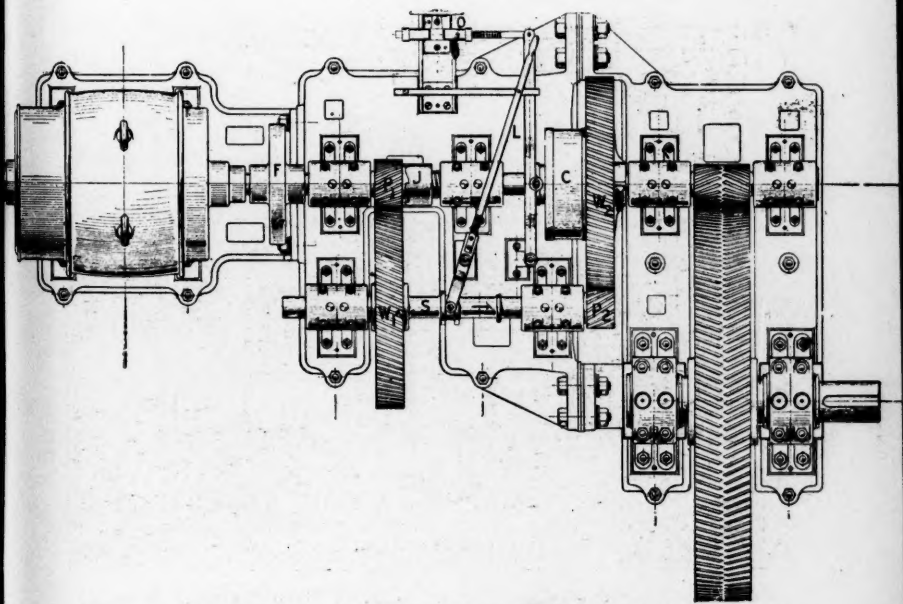

FIG. 33.—SELF-CONTAINED COLD-ROLLING STARTING AND ROLL-TURNING UNIT.
THE POWER PLANT CO., LTD.

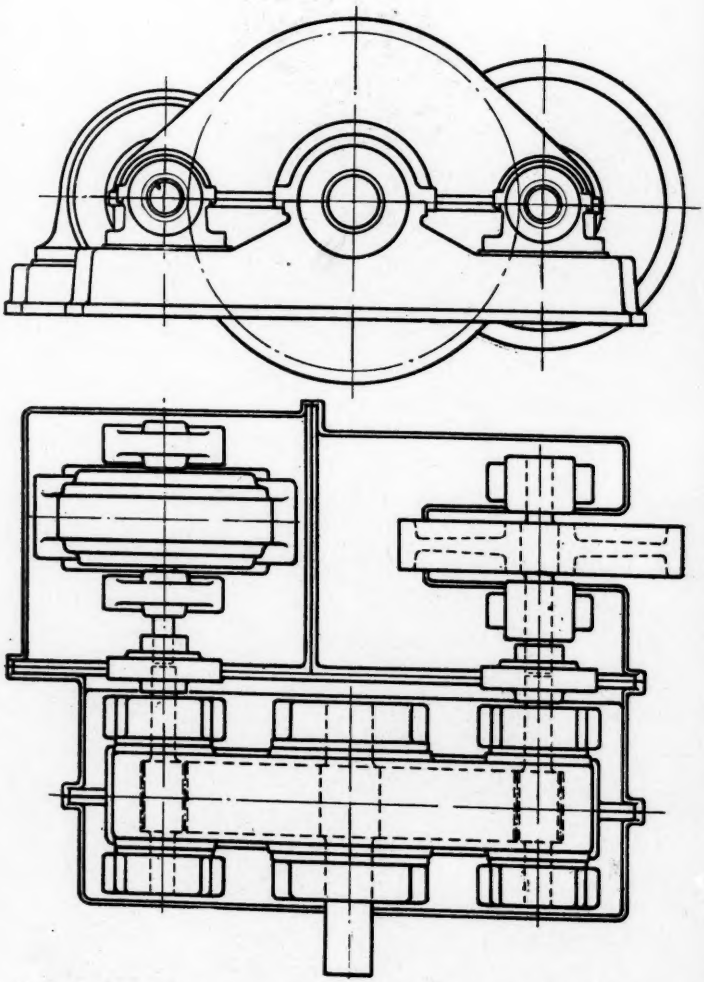

FIG. 34.—ARRANGEMENT FOR A ROLLING MILL WITH HIGH-SPEED FLYWHEEL.
THE POWER PLANT CO., LTD.

tooth going out of gear and the next tooth coming in, so that the load on a straight tooth is suddenly applied, whilst with double helical teeth the full load comes on each tooth gradually.

For heavy rolling mill or similar drives double helical gears are far superior to straight-tooth gears, as their additional strength, freedom from shocks, and ability to carry heavy overloads without breakdown make it possible to use gears of reasonable dimensions for the heaviest loads.

There are three well-known methods of cutting double helical gears—hobbing, planing and end-milling, the first two being the most satisfactory owing to the greater accuracy which can be obtained. The hobbing process, as its name implies, is one by which the teeth are hobbed out of the solid metal by means of hobs, and this is the method adopted for cutting double helical staggered teeth of the type shown in Fig. 7, which shows a high-ratio haulage gear. This method ensures great accuracy of pitch, and all gears with any number of teeth cut with the same hobs will gear together accurately. For cutting by the planing process there are two distinct types of machines; in one the teeth are generated from a rack form of cutter, and in the other from a cutter in the form of a pinion, which is geared with the wheel blank to be cut in the correct ratio. The end-milling process produces teeth of the form shown in Fig. 8. These are cut by means of end mills and, owing to the large amount of metal which a small mill has to remove, the cutters require frequent replacing. Furthermore, it is necessary to have a different form of cutter for each number of teeth in each pitch. It will be readily understood from this that inaccuracies easily creep in in cutting this last type of teeth. Figs. 9, 10 and 11 show wheels being cut by the three processes mentioned.

Fig. 12 illustrates a large reducing-gear unit for a Canadian rolling mill. This unit consists of three reductions, the first two being double helical gears enclosed in a cast-iron casing, and a final reduction with straight-cut teeth. This particular gear was designed for transmitting 200 H.P. with a speed reduction from 600 to 10 r.p.m. The final wheel is 8 feet 6 inches diameter, 3½ inches pitch, 17 inches face, and the total weight of the whole arrangement is, approximately, 22 tons.

Fig. 13 shows a similar unit, transmitting 250 H.P. and reducing from 450 to 2 r.p.m. The large spur wheel is 11 feet diameter, and the whole arrangement weighs, approximately, 30 tons. In both these examples the motor, gear and outboard bearing are mounted on heavy cast-iron base-plates, making a very substantial piece of machinery. The bearings in the casing are of phosphor bronze, and are of the ring-oiling type, the rings dipping into large reservoirs underneath the bearings.

Fig. 14 shows a spur-gear unit for an electrically-driven type rolling mill. The centers of the gears are 45 inches. The unit is designed to transmit 1,500 H.P. in either direction, and the reduction in speed is from 188 : 75 r.p.m. The heavy frame bedplate is of cast-iron, the bearings carrying the shafts being bolted in position. The cover is of light construction, being merely an oil chamber and gear guard, and is so constructed that it may be removed without interfering with the gears or bearings; similarly, any one of the bearings can be removed without disturbing the other parts of the unit. The coupling on the low-speed shaft is forged solid with the shaft and is fitted with cross feather. The lubrication of the gears is by gravity from the tank, the tank being kept supplied by a small pump. The bearings are of the ring-lubricated type, and are also automatically supplied with oil from the same tank. The total weight of this unit is, approximately, 17 tons.

Fig. 15 shows two double helical-gear units, one complete and the other with the top part of cover removed showing the gears, which are of the Chevron type cut by end mills. These units were designed for transmitting 400 H.P. each, reducing from 580 : 180 r.p.m. In these cases also

the bearings are bolted to a rigid framework, and a light cast-iron cover is fitted, acting as a protection for the gears, and also as an oil well. This cover is bolted to the inner ends of the bearings, which have flanges cast on. The bearings are of the ring-oiling type, and no special oiling arrangements are provided for the gears, lubrication depending on the oil carried in the teeth of the gear from the oil in the casing.

Fig. 16 shows a double reduction-gear unit designed for use with a rolling mill. This unit has the drive arranged vertically, the motive power being applied to the bottom shaft, which is underneath the framing. One of these units is placed between each pair of rolls, and is designed for transmitting 50 H.P. reducing from 480 : 36 r.p.m. An oiling arrangement is fitted which ensures ample lubrication to each bearing and to the gears.

Fig. 17 shows a single reduction gear unit of good design for transmitting 250 H.P. and reducing from 400 : 40 r.p.m. The frame is of massive proportions, the low-speed bearings being cast solid with it, ensuring perfect rigidity where it is most needed. The high-speed bearings are bolted and dowelled to the frame; these are made separately to facilitate adjustment for various sizes of gears. All bearings are of the ring-oiling type. A flexible coupling is fitted to the high-speed shaft and a solid flange coupling to the low-speed shaft.

Fig. 18 is an interesting example of a small high-speed increasing gear, to transmit 12 to 24 H.P., and increasing speed from 475 to 3,000 r.p.m. The bearings are of the ring-oiling type, and the gears are lubricated from the oil in the casing by splash.

Fig. 19 shows another speed-increasing gear of the same type connecting an oil engine to a centrifugal pump and transmitting 25 H.P. at 600 to 1,580 r.p.m. This unit is of small proportions, and has a very neat appearance.

Fig. 20 is an illustration of a totally enclosed double reduction gear unit, with high-speed and low-speed shafts in line with one another. The gear is driving calciners through double helical bevel gear, the total reduction being from 1,000 to 1.5 r.p.m. An important point to note in connection with double helical bevels is that there is no end thrust along either shaft, and for this reason, despite the initial relatively higher cost compared with straight-cut bevels, they are being largely used by engineers who study economy in running costs.

So far we have generally considered electric motors as the prime movers, but of recent years steam turbines have come into more general use. Owing to the high speeds at which these invariably run, it is necessary to have some form of reduction gear to reduce their speed to a reasonable figure for certain industrial applications. It is much more economical and efficient to have a turbine running at its best speed in conjunction with an efficient reducing-gear unit than to make a turbine run at a lower speed. For example, the most efficient speeds for turbines for ship propulsion are much higher than the most efficient speeds for propellers, and it has been proved that it is far more economical to employ a gear unit between the turbine and the propeller than to design a turbine to run at a suitable speed for the propeller.

Only an expert who has had considerable experience can design a really satisfactory unit for such conditions. Problems involving most suitable gear for high velocity, design of bearings and casing, and methods of lubrication should not be tackled by the ordinary engineer or draughtsman, as it is not to be expected that they will realize the difficulties. It is here that double helical gears play a most important part owing to their smooth running, due to the continuity of engagement of the teeth reducing the noise made by the gears to a minimum, and at such high speeds the noise can be very great with an incorrect design of gear unit. The important points to be watched are correct design, rigidity,

balance of moving parts, the last being very important and not always given proper attention. The methods of lubrication also call for expert knowledge. High-speed gears must be most carefully designed, and each stage of their manufacture minutely supervised and checked, and the materials for the construction of each part must be carefully selected.

The pinions are usually made from special forged steel, and the wheels from forged-steel rims shrunk on to steel or cast-iron centers and carefully balanced. The bearings are lined with a special brand of anti-friction metal most suitable for the high speeds met with, also oil must be supplied under pressure both to the bearings and the teeth of the gear at the point where they come into engagement, and for a reversing gear the oil supply must be so designed that it can be used when the gear is running in either direction. The oil after being used must be passed through a cooler before going through the pump again.

Fig. 21 shows a typical design of turbine gear unit. The two gear shafts are placed horizontally, and in order to accommodate the wide face widths of the gears without having any whip in the pinion shaft, an intermediate bearing is fitted between the two halves of the pinion. It will be noticed that the teeth are cut with opposite hands of spirals so that the end thrusts set up by the teeth will neutralize each other, and whilst the wheel is prevented from moving sideways the pinion is self-locating in the teeth, so giving a uniform load on each half of the wheel.

Turbine gears are made with a relatively fine pitch, usually under 1 inch, and of great face width. This conduces to a very smooth gliding into and out of engagement, giving a quiet running pair of gears. In all cases of turbine gears it is necessary to use some form of flexible coupling in order to completely isolate the gears from any external shocks or stresses.

Fig. 22 shows a reversible turbine gear with the pinion rotating at 13,000 r.p.m., whilst the wheel makes 2,800 r.p.m. In this case the bearings are exceptionally long and, although the pinion shaft is only $\frac{5}{8}$ inch diameter, the circumferential speed is over 2,000 feet per minute.

If not carefully attended to, a great deal of power can be lost due to faulty lubrication. Unlike slower speed units, the gear must not dip into the oil well or the friction and churning of the oil alone will run away with considerable power and cause loss of efficiency. Consequently, it is necessary in turbine gears to spray the oil under pressure directly on to the gear teeth.

Fig. 23 shows a pair of turbine gears designed to transmit 750 H.P. reducing from 3,600 to 750 r.p.m. The pinion is made of a special steel, and the wheel with forged-steel rim is shrunk on to a cast-iron center. The centers of the shafts are 21 inches apart, and the face width of the gears 20 inches over all. Gears of this description can be generated either by means of hobs or by the planing process. This particular pair of gears work under actual conditions at full load with a constant temperature of 160 degrees F., which will give some idea of the high efficiency obtained, which is due to proper care and attention having been given to their manufacture. The noise made by this pair of gears was quite inaudible at a distance of about 4 feet from the casing.

Another example of turbine gears is shown in Fig. 24. This unit was made for driving a centrifugal pump from a turbine, and is transmitting 120 H.P. at 2,700 to 800 r.p.m. The gear is of very small dimensions, having only 10-inch centers. In this case a center bearing is provided between the two halves of the pinion.

Fig. 25 shows a large turbine gear for ship propulsion. This is transmitting 3,350 H.P. and reducing from 1,450 to 85 r.p.m. This shows very clearly the large reduction which can be obtained with this type of gear. This particular gear is cut by means of hobs and quite a large number have been made.

Fig. 25 is another turbine gear unit transmitting 900 H.P. and reducing

from 3,400 to 600 r.p.m. This is an example of a very good design and the photograph shows quite clearly the forced-lubrication arrangements. This is a similar gear to Fig. 23, except that it is of much smaller dimensions and no center bearing is necessary for the pinion. Similar gears are being extensively used for mill drives and in many cases for increasing the speed of a steam engine to that necessary for driving an electric generator. In driving mills great economy is often effected by this system; instead of having a lot of shafts, etc., an electric generator is driven from the main engine and the various departments independently driven by electric motors.

We will now give our consideration to what may be regarded as the oldest type of reduction gear where larger ratios are required. This is the worm gear. Up till quite recently worm gear was considered to be very inefficient both by makers and users, but owing to the very great improvements in the method of manufacture, and the increase of knowledge which has been obtained of correct design, fairly efficient worm gears are now manufactured. In some cases the worm gear is the only possible method of driving, as, for example, where the wheel is placed in the middle of a long shaft and the motor underneath. This type of gear runs practically noiselessly, which, perhaps, in some cases is of more importance than efficiency. Very large reductions can also be obtained with this type of gear; such reductions as 100 or 130 to 1 being readily obtained, but in such cases the efficiency is necessarily very low. They are also very useful in cases where a non-reversing drive is necessary, such as a hoist.

Fig. 27 shows a worm-gear unit with portions of the casings cut away to show the internal arrangements. This particular gear is fitted with ball journal bearings to both shafts and a double ball thrust bearing to the worm shaft. This latter is a necessity where anything like high efficiency is required.

Fig. 28 is an illustration of a worm-gear unit with a vertical worm shaft and horizontal wheel shaft; the thrust bearing for the worm is on the top of the casing. The lubrication of the bearing is by means of the grease lubricators shown, and the gears are lubricated by splash. A stuffing box is fitted to the lower vertical shaft bearing to prevent leakage of oil down the shaft. Several of these have been satisfactorily used for connecting water turbines to line shafts.

Fig. 29 is an interesting example of a worm gear used for starting a large motor; the pawls engaging with the ratchet wheel being thrown out, and held out of gear by centrifugal force as soon as the large motor gets up to speed.

Fig. 30 is a special arrangement of two worm gears with right and left-hand worms driven from one motor, for driving a shale breaker.

Fig. 31 shows a very large double worm gear with two wheels side by side driven by two worms which are connected together by spur gears, one worm shaft receiving the power from the gas engine and transmitting half the power to the wheel with which it engages, the other half through the other worm to its wheel. This unit is transmitting 165 H.P. with the gas engine running at 170 r.p.m. to a pump; the ratio of reduction being 7¼ to 1. The centers between worm and wheel are 40 inch.

Fig. 32 is an example of a large gear unit for driving a bar-rolling mill transmitting 1,800 maximum H.P. at 162 to 50 r.p.m., the center distance being 54 inches and the gear 20 inches wide; the total weight of the unit being over 20 tons. In the illustration the top cover for the gear has been removed. The gears are lubricated by splash and the bearings are of the ring-oiling type.

Fig. 33 shows a self-contained cold-rolling starting and roll-turning unit which only requires to be coupled up to the motor on one side and the mill on the other side.

The total reduction from motor speed to roll speed is obtained in one reduction of double helical gear with staggered teeth, giving the greatest simplicity and highest possible efficiency. The motor is used for starting the rolls through a back gear, which reduces the roll speed, at full motor speed, so far that the normal motor torque is amply sufficient for starting. The back gear disengages automatically (patent) when the through drive from motor to main gear is put into operation. The back gear also serves for the necessary low speed for turning the rolls, and the whole arrangement is simple and reliable in action.

Fig. 34 is another ingenious patented arrangement for a rolling mill with high-speed flywheel; in this unit the power from the motor is transmitted to the wheel on one side and the power from the flywheel at the other side, and the gears are very much smaller than they would be if the flywheel where mounted between the motor and the pinion and only one pinion used. Both motor and flywheel are connected to their pinions by flexible couplings and the whole arrangement is on a rigid base plate.

The examples described do not anything like cover the complete field on which gear units can be used. They are, however, fairly typical of good practice. Many special adaptations are constantly being made which pass into standard articles, and a word of advice to designers of plant who are considering the use of gear drives is to give the manufacturers who specialize in gearing the fullest possible particulars of the drive and let them have a free hand in the designing. This is not only conducive to the best results being obtained, but in a great many cases standard articles which have proved satisfactory can be supplied at a saving in cost and time.—"Cassier's Engineering Monthly."

VACUUM AND ITS EFFECTS ON STEAM TURBINES.

In all turbine installations it is generally conceded that for economical working, condensing plant should be provided for giving a very high degree of vacuum, say from 28½ to 29 inches, on a barometer of 30 inches. For land purposes, in which the space available for the plant is, to a certain extent, unlimited, and where the power units are relatively small, no great difficulty has been experienced in designing turbines to utilize economically a high degree of vacuum at full load. In marine work, however, in which the space allowed for the machinery is somewhat limited, engineers usually have to be satisfied with a lower vacuum than the above-stated figures, as it is not possible in some cases, especially in units of large powers and limited dimensions, to design turbines to deal economically with the increased volume of steam due to higher vacua.

In turbines of the marine direct-driven type in which a compromise has to be made between the low revolutions necessary for good propeller efficiency and the high blade speed required for efficient working of turbines, the space available generally permits of a turbine of a moderate diameter being used, and, with an exhaust blade length made suitable for the chosen diameter, a pressure drop over the blading may be obtained for passing the requisite amount of steam which will give a fairly high velocity ratio with the chosen vacuum.

The velocity ratio is the ratio between the mean speed of the blades and the circumferential component velocity of the steam through the blades, and, for maximum efficiency, should be about .5 for impulse turbines and about 1.0 for reaction turbines. In actual practice, however, a considerably lower figure than the above has to be chosen for exhaust blades, otherwise the length of these blades would become excessive. The blade length should not exceed about 20 per cent. of the mean diameter of turbine, in order that the losses which occur in the blading due to their radial spread may not become excessive.

In marine-turbine installations of the high-speed type, in which mechanically or electrically-geared slow-running high-efficient propellers are used, the turbines are usually made of a small diameter, and run at a high peripheral speed. In these turbines of smaller diameter, with their limiting blade length, the pressure drop over the blading for passing the required amount of steam will, of course, be much greater than in turbines of larger diameter, and it is seen that in order to obtain the same velocity ratio and approximately equal efficiency of blading as in the larger turbines, a very high peripheral speed would be required. This high peripheral speed has, however, to be limited, otherwise the centrifugal stresses on the moving parts of the rotor will become excessive.

Owing to the limitations in dimensions it is impossible in some cases, especially in units of large powers, to obtain sufficient area through the exhaust blades to maintain a moderate velocity ratio except very flat and inefficient blades be used. A turbine is said to be bladed for a certain vacuum when the velocity ratio at the exhaust end for the chosen vacuum is equal to the mean velocity ratio throughout the turbine. The capacity of a turbine for responding to high vacuum depends on the vacuum for which the turbine has been bladed, and the higher the working vacuum is above that for which the blading has been designed the less is the proportionate gain for the heat available to the turbine. For instance, if a turbine be bladed for a 27-inch vacuum, and is made to run on a 28-inch vacuum, the increase in thermal units available to the turbine, assuming an initial steam pressure of about 200 pounds per square inch, will be about 6 per cent., but the gain in power with the same quantity of steam flowing through the turbine will not be 6 per cent. but somewhere about half this quantity, depending on the design and type of turbine. If the dimensions of the turbines would allow suitable blading for the 28-inch vacuum, then the full amount of 6 per cent. gain in power would be realized. The best vacuum for any turbine to be bladed to is the highest possible vacuum, and suitable condensing plant should be provided for attaining a somewhat higher vacuum than that for which turbine is bladed, but in some cases where the area of exhaust is restricted a moderate condensing plant would seem to be sufficient, otherwise the increase in the steam consumption of the larger auxiliaries required for the higher vacuum, together with the lower temperature of condensate, will tend to reduce to a minimum any apparent gain.

It may be mentioned, however, that any turbine which is bladed for a comparatively low vacuum at full load is quite suitable for dealing with higher vacuum at low loads, and if economy is a consideration at the lower loads, then it is advisable to provide a condensing plant for the higher vacuum, when trials of plant can decide the best vacuum to work at for the higher and maximum loads.—"Shipbuilding and Shipping Record."

LARGE DRY DOCK TO BE BUILT.

The Morse Dry Dock and Repair Company has plans complete for building the largest sectional floating dry dock in the country at its yard at the foot of Fifty-eighth Street, South Brooklyn. This dock will be capable of taking out our largest Dreadnoughts, also merchant vessels of the size of the *Mauretania* or *Vaterland*. According to the plans which have been inspected by the Department of Docks and Ferries, the big floating dry dock will be capable of lifting and lowering ships so rapidly that four vessels can be placed within the dock in twenty-four hours. It is expected that a ship can be lifted in twenty minutes and floated in seven. The Morse Company also has preliminary work under way for the erection of a factory and office building with entrance to the dock yards at Fifty-sixth Street, to be built of concrete and brick.

RESULTS OF MODEL-TANK EXPERIMENTS TO DETERMINE THE ACTION OF A SHIP BRAKE.*

This paper is presented with the idea of showing the interesting data obtained in the model tank at the Navy Yard, Washington, for use in determining whether or not it was feasible to design and install a brake on the side of a merchant vessel to assist in stopping a steamer when in danger of collision or grounding.

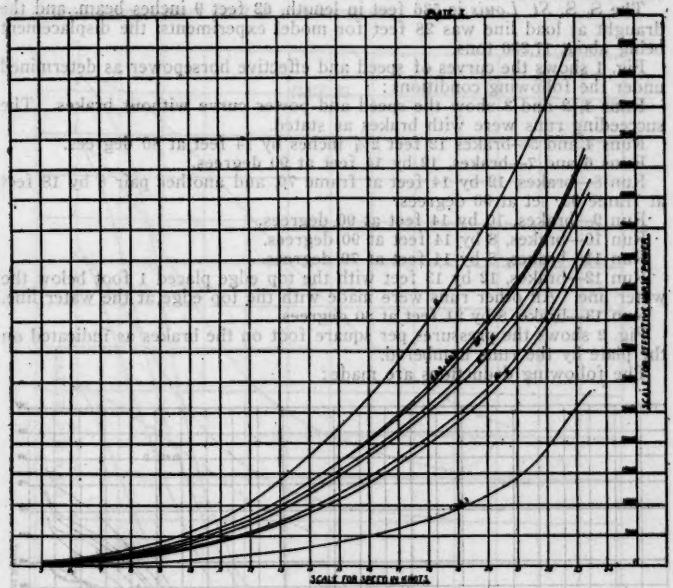

FIG. 1.—CURVES OF SPEED AND EFFECTIVE HORSEPOWER.

The question of installing such a brake is a financial one and is dependent upon the lessened insurance demanded and the advertisement of such an appliance to increase the passenger list.

This question is not one for the naval architect or marine engineer to solve, and is not brought up for discussion on that point.

In the spring of 1910, a set of trials was made on the U. S. S. *Indiana* off Delaware Breakwater to determine the effect of the Lacoste ship brake. Due to a lack of preliminary investigation, this trial showed very little either for or against the brake, and it was some time after that the writer was requested by personal friends to take up the study with a view to designing a practical brake.

After the loss of the *Titanic* interest was awakened to such an extent that a syndicate was formed and the permission of the Navy Department was obtained to make a series of trials in the model tank, and the results

*Paper read by Captain Wm. Strother Smith, U. S. N., Member, at the 23d general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 18 and 19.

of these trials are now presented through the courtesy of the Lacoste Shipbrake Syndicate.

The Wm. Cramp & Sons' Ship & Engine Building Company was kind enough to furnish the lines of the S. S. *St. Louis*, and a model of this vessel was constructed.

Three runs (Nos. 1, 2 and 3) were made to determine the curve of speed and power before fitting any brakes, and the successive 10 runs were made with different areas and settings of the brake to determine the resistances and pressures per square foot.

The S. S. *St. Louis* is 536 feet in length, 62 feet 9 inches beam, and the draught at load line was 28 feet for model experiments, the displacement being about 17,230 tons.

Fig. 1 shows the curves of speed and effective horsepower as determined under the following conditions:

Runs 1, 2 and 3 show the speed and power curve without brakes. The succeeding runs were with brakes as stated.

Runs 4 and 5—brakes 12 feet 2¼ inches by 14 feet at 80 degrees.

Runs 6 and 7—brakes, 12 by 14 feet at 90 degrees.

Run 8—brakes, 12 by 14 feet at frame 76, and another pair 8 by 18 feet at frame 95, set at 90 degrees.

Run 9—brakes, 10 by 14 feet at 90 degrees.

Run 10—brakes, 8 by 14 feet at 90 degrees.

Run 11—brakes, 8 by 14 feet at 70 degrees.

Run 12—brakes, 12 by 13 feet with the top edge placed 1 foot below the water line. All other runs were made with the top edge at the water line.

Run 13—brake, 8 by 21 feet at 80 degrees.

Fig. 2 shows the pressures per square foot on the brakes as indicated on the plate by the runs numbered.

The following deductions are made:

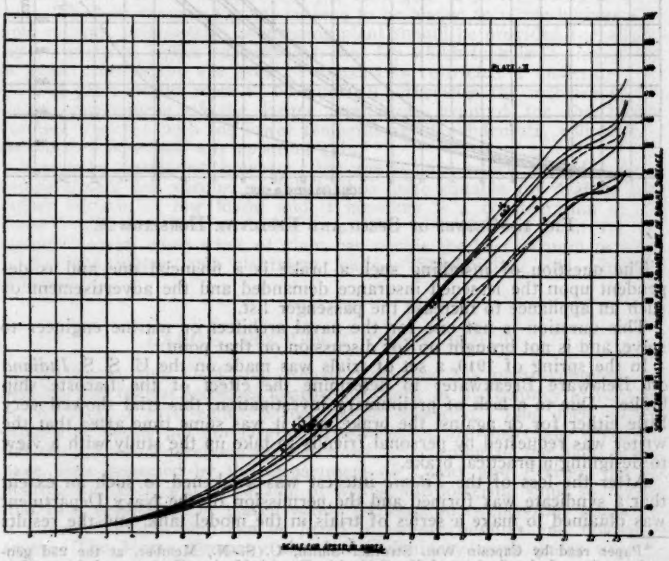

FIG. 2.—PRESSURES PER SQUARE FOOT ON THE BRAKES.

The stopping power varies directly as the projected area opposed at 90 degrees to the fore-and-aft line.

One pair of brakes is more efficient than two, as shown by run No. 8, where the combined area is 624 square feet for the two brakes, the stopping power or resistance opposed is not in the same proportion as the pair of 12 by 14 feet, giving 168 square feet.

The pressure per square foot practically decreases as the width of the brake is reduced and, by plotting curves of pressures at various speeds for different widths of brake, it is found that after a certain width is passed the pressure per square foot is not increased.

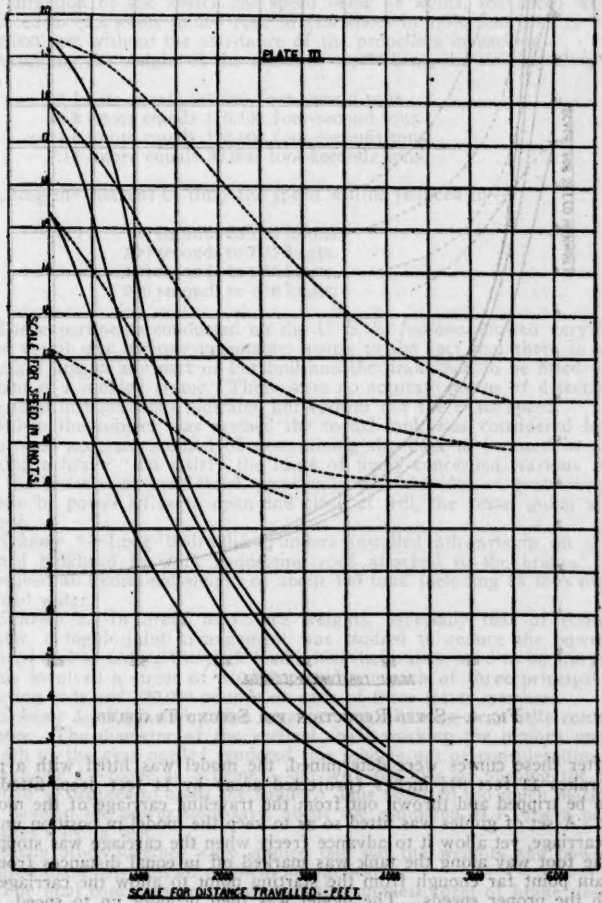

FIG. 3.—SPEED REDUCTION PER FOOT TRAVELED.

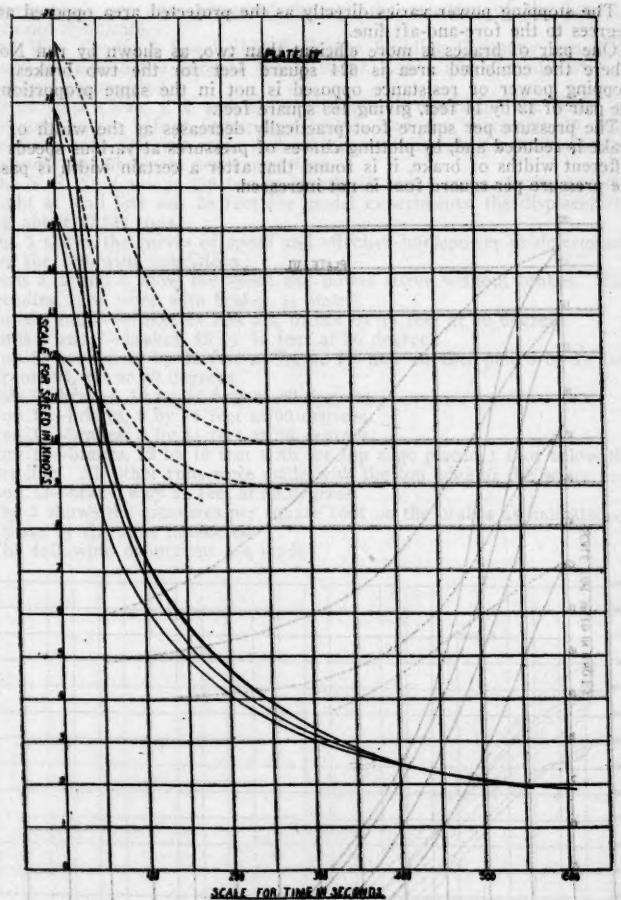

FIG. 4.—SPEED REDUCTION PER SECOND TRAVELED.

After these curves were determined, the model was fitted with a pair of brakes 11 feet 8¾ inches (projected area) by 14 feet deep, fitted so as to be tripped and thrown out from the traveling carriage of the model tank. A set of guides was fitted so as to keep the model in position under the carriage, yet allow it to advance freely when the carriage was stopped.

The foot way along the tank was marked off in equal distances from a certain point far enough from the starting point to allow the carriage to reach the proper speeds. The model was then brought up to speed, the carriage stopped at the determined point, the model released with the

brakes thrown out, and the times marked by stop watches at each marked distance until the model was brought to rest or the limit of the tank reached.

For each speed the model was released with the brakes closed, so a comparison was obtained and shown on Fig. 3 of speed reduction per foot traveled and on Fig. 4 the speed reduction per second traveled.

As a matter of interesting data, the table shows the instantaneous stopping effect of a pair of brakes of different sizes compiled from the curves shown on Fig. 1.

Using a pair of brakes 11.73 feet wide and 14 feet deep set normal to the direction of the vessel, the speed being 18 knots, the speed will be reduced to 15.2 knots in 500 feet, to 11.6 knots in 1,000 feet, to 7.15 knots in 2,000 feet without the assistance of the propellers in backing.

Assuming the weight of the vessel at 17,230 tons, the striking blow at—

18 knots equals 247,300 foot-second tons.
15.2 knots equals 176,300 foot-second tons.
11.6 knots equals 102,700 foot-second tons.
7.15 knots equals 39,000 foot-seconds tons.

Using the element of time, the speed will be reduced in—

50 seconds to 11.1 knots.
100 seconds to 7.35 knots.
150 seconds to 5.85 knots.
200 seconds to 4.70 knots.

The experiments conducted on the U. S. S. *Indiana* showed very little. The vessel was entirely unsuitable, owing to the fact that there is not a straight line in any part of her hull and the brake had to be fitted on by means of a molded frame. There were no accurate means of determining the total instantaneous indicated horsepower nor the exact speed.

When the subject was revised the model tank was considered by far the most accurate method of determining the data to be used in designating a brake. To satisfy the ideas of many concerned, various forms of the brake were considered, principally with the idea of controlling the brake by power so as to open and close at will, the vessel going at full speed.

Scheme 1.—Long hydraulic cylinders installed athwartship on a stiffened bulkhead to work connecting rods attached to the brakes. This involved an additional weight of about 190 tons, including 33 tons of contained water.

Scheme 2.—In order to reduce weights, especially that of contained water, a toggle-joint arrangement was studied to reduce the power required and to reduce the peak load when the brakes were to be started in. This involved a stress of 340,000 pounds on each of three principal connecting rods and 320,000 pounds on each of three lesser members.

Scheme 3.—Rack and pinion arrangement, allowing very little contained water. The diameter of the vertical shaft working the pinions and the width of the gear needed rendered this scheme out of consideration.

Scheme 4.—Same as Scheme 3, but with the idea of pushing the brake out to 35 degrees to assist the rudder. This required too much machinery to develop the power necessary, about 500 estimated H.P. for a pair of brakes 9 by 13 feet.

Many other schemes were studied out, but the weights and power required made any and all of them impracticable.

The final conclusion reached—that the simplest form of brake designed to be set in the side of a vessel, to be released instantly by the officer on the bridge by throwing a lever and opened by the pressure of the water

without further thought; to be supplied with only enough power and mechanism to haul in the brake when the vessel was stopped or going astern and lock it fast for future use—was the only practical solution of the question.

TABLE SHOWING EFFECT OF BRAKES.

Size of brake.		Stopping effect at various speeds in effective horsepower speeds.								
Wide	Deep	15	16	17	18	19	20	21	22	23
Ft.	Ft.									
12	14	13,000	15.500	18,250	21,750	23,250	28,500	32,200	35,250	38,500
10	14	10,600	12,750	15,000	17,500	20,000	22,500	25,600	28,300	30,400
8	14	7,850	9,500	11,250	13,500	15,500	17,500	19,800	21,900	24,000
6	14	5,300	6,400	7,590	9,100	10,460	11,810	13,360	14,780	16,200
12	13	11,800	14,100	16,600	19,800	21,100	25,900	29,300	32,000	35,000
10	13	9,600	11,600	13,600	15,900	18,000	20,400	23,300	25,750	27,660
8	13	7,100	8,6co	10,200	12,250	14,100	15,900	18,000	19,900	21,800
6	13	4,650	5,650	7,650	8,250	9,500	10,700	12,100	13,400	14,700
10	12	8,600	10,400	12,300	14,300	16,000	18,300	20,900	23,175	24,880
8	12	6,300	7,700	9,100	11,000	12,600	14,300	16,000	17,900	19,600
6	12	3,900	5,050	6,850	7,400	8,500	9,600	10,800	12,000	13,200
10	10	7,000	8,600	10,100	11,900	13,300	15,200	17,000	19,300	20,700
8	10	5,200	6,400	7,500	9,000	10,400	11,800	13,100	14,500	16,200
6	10	3,100	4,100	5,600	6,100	7,000	8,000	10,100	12,000	12,200
8	8	4,100	5,100	5,800	7,000	8,300	9,400	10,400	11,500	13,000
6	8	2,400	3,200	4,300	4,700	6,500	7,400	8,200	9,000	9,500
6	6	1,700	2,200	3,000	3,200	4,600	5,400	6,000	6,600	7,000

With these points in view a type design was made for the Canadian Pacific S. S. *Empress of Asia*, practical, strong and simple, for emergency use only. The officer on the bridge has but to pull a lever and the brake will open and exert its resistance. When the danger is past, it requires but the attention of one man on each side to haul in and secure the brake.

With this done the technicist closes his work and the financier and promoter begins to write his story.—"Shipbuilding and Shipping Record."

FORCED FEED LUBRICATION ABOARD SHIP.

From extensive inquiry, the consensus of opinion is that successful practice in the use of oil comes from unceasing care in handling. Reliable dealers when they sell an oil for a certain purpose are very certain that it will fulfil that purpose and are very glad to aid in any way they can. They have in their employment the best chemists and practical engineers available. They spend thousands of dollars yearly, and their success as oil distributors depends upon the successful operation of the oil they sell. So it is reasonable to assume that when an oil apparently does not do its work the method of application and not the oil is at fault.

Faults may be inherent of the installation or, what is more probable, of the care and handling. The object of the installation is to (1) supply the requisite amount of oil to the desired bearings in varying amounts as demanded; (2) to supply it at the desired temperature—this includes

heating and cooling apparatus; (3) to allow sufficient storage; (4) to provide a method for purification, the removal of sediment and water, it being safe to assume that no system is so perfect as to discount these two items.

The amount of supply is a matter of pipe and pump design. There is little choice in the pumps on the market, the ordinary plunger type is very good. For small installations the gear pump is excellent. Pipe diameter must be sufficient to allow a generous supply and to keep the velocity below the critical point. Sharp bends must be guarded against. In laying out the piping it will be found of much value to have a slightly curved section in preference to a straight section, so that when setting up sections this curve will allow the joints to be well made. Ground flange faces, while initially expensive, are cheaper in the long run. The suction lines must be above the highest possible level of bilge water, otherwise a leaky joint on the suction line will fill the system with dirty bilge water.

Heating coils are unnecessary on ships. With the engine-room temperature varying very little, it is hard even to imagine a possible condition where they are necessary, except for ships operating in high latitudes. They may aid in settling, but not to such an extent as to authorize their installation on account of the cost, liability of leaks, increased weight and inconvenience in cleaning the tanks. Cooling coils should never be installed in a tank. Use an oil cooler; it is more efficient and leaves the tanks free for cleaning. Many oil tanks are not properly cleaned because of the mass of piping installed in them.

The supply tank must be of sufficient capacity to insure a generous supply. There are many rules. The best practice is to copy a successful installation. The tank should have at least a capacity equal to one-tenth of the hourly supply pumped through the system. Oil can be used over and over for periods varying from six months to a year, if properly filtered and cared for. The tank must be of sufficient size to fill the system and then have an available supply of from 50 to 100 gallons, depending on the size of the installation. In a large installation, as a battleship, there must be at least 100 gallons available. Knowing rate of supply, keep at least two minutes' supply available, as it will take at least this time to shift connections.

The tank must have three connections: (1) supply (this should be at the top); (2) suction (water and sediment settles to the bottom, so this connection must be above the highest possible water level; in a 400-gallon tank this water deposit may run as high as 40 gallons); (3) drain pipe to run water off (this must be at the bottom and lead to the bilges). The matter of cruising supply is often neglected. Oil will keep indefinitely, hence no loss results from deterioration or other loss attendant to keeping from 500 to 1,000 gallons on hand. A small leak will eat up 200 or 300 gallons in less time than it takes to tell about it. As a minimum in a large installation there should be a reserve of at least 500 gallons. This entails an investment of about $75 (15/12/6). Ordinarily, when cruising, the make-up oil should not exceed 2 gallons a day.

To illustrate the necessity of supply I will quote an incident that happened to me several years ago. While in charge of a turbine installation of about 10,000 horsepower, a leak developed in a spring bearing. The result was an hourly loss of 20 gallons. The ship had to keep going, and luckily we were carrying a reserve of 500 gallons, due to previous experience. It might also be stated that this reserve was being carried at considerable inconvenience. Previous to this a sister ship had made port on a mixture of water, gasoline (petrol), pump oil, cylinder oil and graphite, the use of said mixture causing several weeks' work. A reserve of 500 gallons in this case would surely have been a saving. Anyone who has spent a day working around oil lines will appreciate this in all its beauties.

Next comes the filtering system with its many troubles. Oil filters are like the measles, we don't like them but we have to have them, and once you have them they are a constant source of irritation, and unless properly treated leave bad after effects. To begin with, most oil filters are too small. Very much like three deuces, they give you confidence, but you don't brag about them. The oil must have time to filter. If sent through too rapidly the filters are useless. The filter must be cleaned regularly, at least weekly, and oftener if possible. There are examples where the filters have not been opened for months and the operator in the meantime has been yelling about the poor oil the distributors supply, when in reality the oil was excellent and he was only showing up himself. A dirty filter will contaminate the best of oil. If your oil is gradually becoming contaminated, try your filter, and it is a good bet that the trouble is there.

The following is a method that I have found to be good in keeping an eye on the oil. Every three or four days take a pint bottle and fill it from the tank in use. Allow it to settle and the water and sediment will collect at the bottom. Any increase in contamination will be discovered at once.

If you wish to go into the subject still more, the following is recommended. Upon request the distributors will supply to you a neat case containing a sample of each oil that you have in use. When the oil is received, compare it with the standard sample as regards color, odor and general characteristics and satisfy yourself that you have received the proper oil. Now each oil has a characteristic color and odor that after a little practice can be easily recognized. As the oil continues in use the color changes, and, as a rule, the oil gradually becomes darker. To one unfamiliar with oils it may seem to indicate that your oil is going bad, but this is not so. To satisfy this fear try the following method: When the oil is received fill a small bottle, lay it aside and mark this No. 1, with the name of the oil and the date; two weeks later remove another sample, mark this No. 2, noting the date. You will notice that No. 2 is darker than No. 1. Remove about five of these samples and you will notice that Nos. 4 and 5 show little change in color. These five samples will give you a good line on the manner in which the oil should change color and will give you a color scale for reference with future oil of the same kind.

The oil, while darker, should in all cases be clear; if cloudy, the chances are that water is present in excessive quantities. As regards odor, not much can be said. This is a matter of practice and experience except that water gives the oil a stale, rank odor that is easily recognized. The above-mentioned samples are also of great help in the determination of the amount of water and sediment present.

One of the principal sources of contamination aboard ship is foreign oil. Now this sounds foolish, but, believe me, it is very often sad, as mixing a compound oil with a straight oil is very much like using turpentine instead of olive oil. If you don't believe it, just try it—yes, I mean both. The favorite way for this foreign oil to get in the system is via the rod glands. All cylinder oils leave more or less deposit, and as steam blows through, this deposit is blown down the rod and sooner or later finds its way to the crank pit, and from there into the supply tank. This sounds "fishy," but some day, when the rest of the gang is scrubbing the anchor, sit down (though real engineers are not supposed to do this) and figure out how many times your piston rods travel up and down during six months. And if you should have the misfortune to get an oil mixture you will think that Sherman was talking of oil systems and not of war.

Well, to go back to our filters, the data on filtration are limited, and what really good data there are can only be obtained from the dis-

tributors. And let me say right here that they are always willing to help you and in most cases will send a representative upon request. This fact is either little known or often neglected, and I most earnestly advise anyone who has troubles to take them to your distributor, and in this way save time and money and at the same time give their oil a fair show.

Oil troubles aboard ship are many, and to go into detail would take many pages, but there is one more that I would like to mention, and that is water in the oil. Water aboard ship is always a proposition, and we all acknowledge that the captain uses enough daily to supply make-up feed for a week, but when it comes to getting it in the oil system, well, it is plain, ordinary h——. And the worst of it is that you can't keep it out. Its effect on the system is the same as the effect of the mule on the fellow who twisted his tail—hospital. Oil as received will contain in the vicinity of 1 per cent. water. On board ship it will collect from 5 to 10 per cent. water. If it goes over five per cent. an investigation should be started. The best way to nip it in the bud is to use the sample bottles mentioned above. Notice how strong I am for those sample bottles. Just try them and you will be, too. The water and sediment settle to the bottom and any increase will be noticed at once. You can get action before there is anything except water in your system and the journals begin to show pit holes big enough to tap.

What has been said above applies to all ship installations in general. Specific cases require their own analyzing. In experience with various installations, large turbines, high-speed generators, large and small reciprocating engines, the points mentioned have been found to be quite general. In addition there are many other points as regards viscosity, flash point, etc., that come under the chemical end and need not be considered here, but which are nevertheless very important. Oil troubles aboard ship are, in nine cases out of ten, due to water and poor filtering conditions. In other words, out of every ten cases there is one in which the oil may be at fault and nine in which the care and management are at fault.—D.—"International Marine Engineering."

VARIATION OF FRICTIONAL RESISTANCE OF SHIPS WITH CONDITION OF WETTED SURFACE.

By Naval Constructor William McEntee, U. S. N.

ABSTRACT.

Considering that frictional resistance is the most important element in the resistance of practically all ships, it is noteworthy that investigations of conditions affecting it have been relatively few. Although there have been many attempts made to reduce this resistance by different methods and there are many references made to its increase from fouling, there appear to be relatively little actual data as to the quantitative variations obtained. The results here given are based on a series of tests made at the United States Experimental Model Basin during a period extending somewhat over a year.

In considering this general subject two queries naturally arise: (1) On actual ships, what is the increase in resistance due to fouling? (2) Is it possible to reduce the frictional resistance by modifications in the condition of the frictional surface, or by the application of materials which would tend to make the surface smoother?

EFFECT OF FOULING.

The method adopted to investigate this subject was to expose to fouling in sea water twelve 10-pound steel plates 2 feet wide by 10 feet long, painted with two coats of anti-corrosive paint. The plates were placed in the water July 10, 1914, at a point in the Chesapeake Bay near the Navy Yard, Norfolk, Va., where the water ordinarily has a density of 1.02—that is, contains about 75 per cent. as much salt as is ordinarily found in sea water.

TABLE I.—TABLE SHOWING INCREASE IN RESISTANCES OF PLATES WHEN FOUL AS COMPARED WITH THE SAME PLATES CLEANED AND FRESHLY PAINTED.

Months in water. Speed, knots.	Increase in resistance, in pounds.											
	1	2	3	4	5	6	7	8	9	10	11	12
2	0.24	0.40	1.17	2.09	2.50	2.52	2.80	2.76	2.61	2.97	3.09	3.36
3	0.72	0.88	2.83	4.94	5.79	5.72	6.45	6.31	6.17	6.93	7.28	7.78
4	1.48	1.56	5.19	9 09	10.53	10.22	11.79	11.38	11.02	12.75	13.39	14.20
5	2.55	2.45	8.41	14.59	16.71	16 08	18.55	18.03	17.32	20.30	21.40	22.50
6	4.02	3.54	12.42	21.40	24.38	23.20	27.20	26.16	25.13	29.83	32.43	33.07
7	6.77	4.74	17.35	29.60	33.45	31.70	37 00	35.95	34.23	41.15	43.4	44.65
8	7.82	6.20	22.90	.....	44.20	41.60	48.65	47.50	44.80	54.50	56.8	60.10
9	10.40	7.75	29.40	.....	56.35	52.85	.....	.....	56 95	69.85	.....	.....
10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Coef. foul plate	0.01138	0.01282	0.01667	0.0239	.0255	.0252	.02745	.02666	.0275	.0285	.0273	.0292
Exp. foul plate	1.994	1.928	2.029	2.002	2.003	1.988	2.000	2.000	1.967	2 015	2.055	2.035
Coef. painted plate.	0.01071	0.01004	0 01000	0.01189	.0108	.0095	.0108	0.01013	.01075	.00903	.0096	.0095
Exp. painted plate	1.869	1.918	1.937	1.855	1.874	1.938	1.880	1.912	1.869	1.848	1.914	1.924

Coef = Coefficient. Exp = Exponent in formula $R = S f V^n$.

The plates were removed from the water at intervals of one month, carefully packed so as not to remove any of the fouling, which consisted mostly of small barnacles, and shipped to the model basin, where they were tested to obtain the frictional resistance.

Each plate as received at the model basin was tested at speeds ranging from 1½ knots to 8 or 9 knots, and gave results shown in Table I. After this the plate was carefully cleaned, scraped and repainted with a coat of Norfolk anti-fouling paint and retested at the same range of speeds as when fouled, and the coefficient and exponent for the clean condition determined. The fouling matter removed from the plates was weighed so as to give a measure of the amount of fouling.

It will be seen that the maximum increase is such that the resistance when foul is four times as great as when the plate is clean and freshly painted.

Referring to Table I it will be seen that not only does the coefficient of friction increase greatly, but that the exponent in the formula, $R = SfV^n$ also increases and has a value averaging about 2.00 as compared with the value of about 1.88 for the clean plates.

These results indicate the importance of keeping a vessel's bottom clean and well painted, though it is not considered that the relative effect on a ship's resistance due to an equal amount of fouling would actually be as great as that obtained in these experiments, as it seems probable that the decrease in frictional coefficient due to increase of length would be relatively greater for the foul condition than is found to be the case for clean, smooth surfaces.

EFFECT OF LUBRICATION.

At various times attempts have been made to reduce the frictional resistance of ships by lubricating the underwater surface or by polishing it to a high degree of polish with various materials. One favored method used for racing yachts is to polish the bottom with black lead or graphite. An investigation of the benefits to be expected from such polishing or lubrication has recently been made at the model basin with two friction planes, which were roughly 20 feet by 2 feet by ¾ inch, with ends sloped forward at an angle of about 30 degrees. Total wetted surface for each is about 82 square feet. The planes were sandpapered smooth and coated with shellac, and in this condition were each towed in the water at speeds ranging from 1 to 11 knots.

One plane was next coated with black lead or graphite, which was mixed in sour milk and applied with a brush to the shellacked surface of the plane. When dry the lead was rubbed down to a fine, smooth polish. Ten runs were made with this plane at speeds from 5.1 to 6.5 knots. The graphite seemed to cling well to the surface and showed no abrasion or wear. When wet, however, it was easily rubbed off.

The next material tried was ordinary ivory soap, which was mixed as a thick emulsion in water and applied with a brush. The plane was run at speeds from 6 to 9 knots The soap left a whitish streak in the water the first run, but not thereafter. The resistances obtained on all of the runs were consistent, so that the quality of the surface was probably unchanged during the course of the experiments.

Two other lubricating materials experimented with were heavy cylinder oil and light cylinder oil. The oil was applied to the shellacked surface of the plane when dry with a brush, the plane being suspended over the position in which it was attached to the towing carriage, so that it could be run in the water as quickly as possible after the application of the oil. The plane with the heavy engine oil was run at speeds from 6 to 8½ knots. The observations obtained were consistent, but the slope of the resistance line when plotted on log paper indicated that at speeds of about 3 knots the frictional resistance would fall below that of the smooth, shellacked surface. This peculiarity is possibly due to the fact that parts of the oil formed in drops or globules and wavelets, increasing as the speed increased, thus changing the condition of the surface which might, if tested at lower speeds, not have given results lower than shellac.

The experiments with the light engine oil indicated that for the first two runs the resistance was a small percentage greater than for the shellacked surface. Apparently as the oil was washed off the resistance decreased until the resistance agreed with that previously obtained with shellac.

The results of the experiments indicate that no advantage over a smooth varnish or shellacked surface is obtainable with the materials used. For each lubricating surface the resistance is greater than for the shellacked surface. The resistance increases in the following order: Black lead, light engine oil, ivory soap, heavy cylinder oil.

The lubricating surfaces were tried only over a limited range of speeds, from 6 to 9 knots, as it was thought that the exponent in the formula $R = SfV^n$ could not be used, owing to the changing condition of the surface due to the washing off of the lubricating material. Actually it was found, however, that the light engine oil was the only material for which the resistance changed rapidly, and that washed off quickly so that the resistance came down to that for shellac.

In Table II is given a tabular comparison of the results:

TABLE II.—WITH CONDITION OF WETTED SURFACE.

Surface.	Plane.	Area, sq. ft.	Coef. of friction.	Exponent.	Net resistance at 7 knots, in lbs.	Percentage increase in resistance at 7 knots.
Shellac	AB	82.12	.00878	1.883	28.1	.....
Shellac	CD	82.20	.00849	1.886	27.4	.....
Black lead	CD	82.20	.00866	1.886	27.9	2
Light engine oil	AB	82.12	.....	...	28.3	5*
Ivory soap	AB	82.12	.01045	1.898	34.5	23
Heavy cylinder oil	CD	82.20	.00484†	2.38†	40.5	48

* At 6 knots.
† This low coefficient of resistance is combined with a high velocity exponent and probably would become greater at speeds lower than those at which experiments were made.

—"International Marine Engineering."

SHIP CALCULATION, RESISTANCE AND PROPULSION.

During the International Engineering Congress held in San Francisco in September, Rear Admiral D. W. Taylor, Chief Constructor of the United States Navy, gave a most interesting and instructive paper on the above subject. It is possible to refer to only a few of the points discussed. Several years ago he read a paper before the Amercian Society of Naval Architects and Marine Engineers on a method of determining the lines of ships by formula. For some 15 years this has been in use at the United States model basin at Washington, not with the idea that they give lines of minimum resistance, but simply to obtain lines possessing desired shapes. The main features of the forms of ships can be expressed mathematically, and these features can be reproduced in new vessels of different dimensions. By the application of this method considerable time is saved at the Washington tank, where they deal with as many as 150 distinct models in one year. Practically all United States naval vessels designed during the last 10 years have had mathematical lines.

Although it is more than 40 years since Froude built the first model tank in his garden at Torquay, it is only since a comparatively recent date that model tanks and their results have been generally accepted as ordinary tools of the naval architect, thoroughly to be relied upon. Here and there a skeptic may exist today, but the law of comparison as applied by Froude is now generally accepted. At the United States model basin, during the last 15 years, the Froude methods have been applied to the models of some 189 United States vessels, having a total value of £95,000,000. In the cases of two vessels only have the results of the trial of the full-sized ship differed materially from what was to be expected from the model results. These were sister ships with exceptionally full sterns due to which eddies were formed under the quarters with accompanying increase in resistance, which corresponding eddies did not appear in the model.

With regard to a frictional resistance, Rear Admiral Taylor had some

interesting things to say. The frictional resistance is determined from estimates made by Froude 40 years ago. It is very difficult to determine with minute accuracy the actual frictional resistance of a large vessel, the workmanship, condition of plating, and other minor factors affect it appreciably, and marine growth may increase friction radically. Experiments made with plates exposed to fouling indicated that in seven months a marine growth, by no means excessive, would increase the frictional resistance by as much as 210 per cent.

In connection with the friction of models as regards variation with temperature, it was planned to run a model of a battleship and one of a torpedo boat monthly, and note temperature upon each occasion. Sir Archibald Denny has stated that the correction should be 4 per cent. for 10 degrees F., whilst Mr. Baker corrects the results obtained at the Teddington tank by 3 per cent. for the same range of temperature, but only on the skin friction portion of the resistance. With change of water temperature there was a perceptible change in the resistance of both the battleship and destroyer models. For the battleship model, the decrease in resistance for 10 degrees F. increase in temperature was about 1.9 per cent. of the frictional resistance, and for the destroyer model 2.3 per cent. of the frictional resistance. This variation is less than that obtained in the British tanks, but the difference may be explained by the fact that wood models were used in the one case and wax in the others, and by the different qualities of the waters.

Rear Admiral Taylor declares that the results of model experiments have been rather discouraging to the development of any formula for expressing the resistance of a ship. A factor which may be of importance at one speed is almost negligible at another. If two vessels are taken, identical in dimensions and displacement, and one built with fine ends and the other with full ends, the fine-ended vessel will offer much less resistance over a certain range of speeds, while at extreme speeds the full-ended vessel will be materially better. Attempts to devise a formula which would accurately express the plotted results of the standard series contained in Taylor's book, "The Speed and Power of Ships," have not been very successful.

The displacement of a ship is the primary factor in its resistance, and the proportions have a determining influence. Viewing the latter aspect, the most important feature is the length used for a given displacement; beam and draught are comparatively minor factors. Length is of peculiar value for cutting down the wave-making resistance, hence for fast vessels, where wave-making resistance is of such importance the total resistance may nearly always be decreased by increasing the length. A very important factor is the prismatic or longitudinal coefficient. For speeds up to $.8 \frac{V}{\sqrt{L}}$, the large midship section and fine ends are nearly always favorable to speed. About a speed length ratio of .95 variation in prismatic coefficient does not produce much effect. For high speeds, however, above, say, $1.23 \frac{V}{\sqrt{L}}$, a reduction in resistance is obtained by using full longitudinal coefficients.

The fore body has greater influence upon the resistance than the after body, and there is no doubt that fine or hollow lines forward near the surface, are favorable to speed. Many seagoing people and naval architects are opposed to hollow waterlines forward, but there are indications that the results of model-tank experiments are making more and more impression upon practical people.—"Shipbuilding and Shipping Record."

PREVENTION OF SPONTANEOUS COMBUSTION.

By Arthur Haver, Jr.

Occasionally fires break out at sea, of their own accord. A coal cargo is especially dangerous in this respect Spontaneous combustion is supposed to be caused by extremely minute particles of coal dust or other inflammable matter floating in air of the hold of a ship. The particles jostle and clash against one another and the friction thus caused raises their temperature towards ignition point. Moreover, the particles are well aerated with the air in which they float, and their oxidation is consequently rapid. By oxidizing, the particles become hotter, over and above the heat brought about by the friction of colliding with one another. In this way ignition point is reached, the particles take fire, and, by igniting larger particles, eventually set fire to the coal itself. In order to provide against spontaneous combustion Lloyd's underwriters require perfect ventilation of cargo. By means of ventilation the hold is cooled to some extent.

Ventilation, however, is objectionable on the ground that a plentiful supply of air facilitates oxidation. Now, when a substance oxides heat is generated. Therefore, by assisting oxidation ventilation raises the temperature of the particles of coal dust towards ignition point. In this way, ventilation may actually help to bring about fire by spontaneous combustion. It is possible that ventilation defeats its own object under certain circumstances.

The following experiment will show how ventilation may place a ship in extreme danger from fire by spontaneous combustion: Let some iron filings be dissolved in a solution of carbon-bi-sulphide. Now, iron, as everybody knows, cannot be ignited by a lighted match. It requires the temperature of a furnace to make iron burn. The iron filings quickly dissolve. Expose the solution to the air by pouring it into an open, shallow vessel such as a saucer. The carbon-bi-sulphide is very volatile and quickly evaporates, and leaves behind the iron as a powder consisting of extremely minute particles. On the instant that the iron particles become dry—and, therefore, exposed to the oxygen of the atmosphere—they burst into flame and are quickly consumed. The experiment may be repeated and the same result obtained by dissolving sulphur in carbon-bi-sulphide.

Now, since iron can be made to burn furiously without applying any heat to it, simply by reducing it to a sufficiently fine powder and exposing it to the air, how much more dangerous is coal dust, which is more inflammable than iron, especially when the coal dust is oxidized, and blazed, and heated up by the plentiful ventilation required by Lloyd's underwriters. A coal cargo is said to give out heat. On this account, it is thought desirable to ventilate coal cargoes very effectively in order that the heated air of the hold may be replaced by cool air, thus retarding oxidation and preventing spontaneous combustion. The writer suggests that although ventilation is cooling it has the disadvantage of promoting oxidation and thus, while the cargo is cooled physically, it is heated chemically. The writer thinks, therefore, that an outbreak of fire at sea may be brought about by the very precautions that are taken to safeguard against it.

To show that ventilation may possibly heat up rather than cool a strongly oxidizable cargo, such as coal, let us borrow from the blacksmith some ideas regarding combustion.

The blacksmith, by means of his bellows, blows a blast of cold air into his fire. At first sight, one would naturally suppose that the cold air would cool the fire and prevent it from burning so brightly. As a matter of fact, the blast has the contrary effect, the fire becomes hotter instead of cooler, because it receives a larger supply of oxygen in a given time.

In the same way, one may reasonably suppose that ventilation by promoting oxidation would heat a coal cargo rather than cool it. It should be noted that oxidation and burning represent the same chemical action. Oxidation is confined to atoms and molecules, and may therefore be regarded as fire on a small scale. It should also be noted that fire may break out at any low temperature provided that the particles of inflammable matter are sufficiently minute.

The writer thinks, therefore, that the ventilation of a coal cargo is of very doubtful value, and suggests that other means should be employed to provide against danger of fire by spontaneous combustion. A very effective way of accomplishing the end in view would consist in replacing the air in the hold by a different gas such as would not support oxidation and combustion.

Steam is a cheap gas, and conveniently at hand for this purpose. All that is required is to dispense with ventilators and connect the donkey boiler with the hold spaces by means of a steam pipe communicating with each hold; an outlet cock should be provided for each hold so as to allow the air to escape.

When the loading of a hold is completed and the hatch battened down let the steam be turned on and the outlet cock opened. As the steam enters the hold the air in the hold is displaced and escapes by the outlet cock. After about 20 minutes let the steam be shut off and the outlet cock closed. The atmosphere of the hold now consists of (say) half as much air as the hold originally contained, the other half being water vapor. According as the steam condenses the pressure in the hold becomes reduced. After an hour let a little more steam be injected into the hold but without opening the outlet cock so that the pressure in the hold may be brought up to 15 pounds per square inch to balance that of the atmosphere. Repeat the same operation with each hold.

Now, spontaneous combustion cannot take place in a hold in which there is a great deal of dampness, insufficient air and no ventilation. Dampness will not harm a coal cargo.

A shower of rain, as everybody knows, purifies the air of our cities by laying the dust. In a manner similar to rain the steam injected into the hold will clear the air of the hold and, on condensing, will lay those dangerous particles of coal dust which cause spontaneous combustion.

The outlet cock should have a diameter several times that of the steam pipe, in inverse ratio of the atmospheric pressure of the air escaping from the hold to the boiler pressure of the steam entering the hold, in order that the weight of the injected steam may be equal to the weight of the ejected air. Lloyd's underwriters will not allow hatch battens to be secured by anything more than their own weight. The idea is to allow the hatch to blow off, thus acting as a safety valve, when gases accumulate in the hold.

The writer thinks that the hatches should be firmly secured so as not to blow off and that proper safety valves should be fitted to each hold to blow off at 7 or 8 pounds per square inch.

Then, if the ship should strike a rock, let steam be injected into the hold. The pressure of air and steam will prevent the sea water from rising in the hold, thus preserving the ship's buoyancy till she can be got into dry dock.—"The Steamship."

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1916.		Percentage hull completed Feb. 1, 1916.	
						Jan. 1	Feb. 1	Total.	On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	98.99	99.29	99.0	99.0
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	97.00	98.33	99.2	99.2
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	89.27	93.54	96.5	96.5
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	67.63	71 98	81.5	80.0
40	*California*	Navy Yard, N. Y.	Electric	4	21	4.00	10.00	20.0	10 7
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	19.12	23.71	37.9	14.9
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	35.99	39.17	53 0	48.0
	DESTROYERS:								
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	95.10	95.79	89.8	89.7
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	91.05	93.91	89.9	89.8
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr .	2	29.5	98.66	99.37	99.9	99.9
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	95 82	96.73	95.3	95.3
63	*Sampson*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	77.24	82.92	75.3	73.4
64	*Rowan*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	75.33	79.07	70.7	66.0
65	*Davis*	Bath Iron Works	Pars. trb. grd. cr.	2	30	70.32	77.37	69.2	67.2
66	*Allen*	Bath Iron Works	Pars. trb. grd. cr.	2	30	69.81	76.90	67.7	65.5
67	*Wilkes*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	55.85	58.13	52.0	49.3
68	*Shaw*	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	28.10	41.68	18.9	13.3
69	.....	Navy Yard, Mare Isl'd	.....	2	32	....	1.2	....	....
70	.....	Navy Yard, Norfolk.	.....	2	32	....	....	....	....
71	.....	Seattle Con. & D. D. Co.	Pars. trb. grd. cr.	3	30	....	....	....	....
72	.....	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	....	....	....	....
73	.....	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	....	....	....	....
74	.....	Bath Iron Works	Pars. trb. gearing	2	32	....	....	....	....
	FUEL SHIPS:								
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	84.93	88.40	99.1	99.1
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	9.82	14.87	29.3	20.3
	DESTROYER TENDER:								
	Transport	Navy Yard, Phila.	Reciprocating	2	14	28.70	32.00	47.8	46.5
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	21.00	22.00	42.9	41.3
	TUGS:								
17	*Wando*	Navy Yard, Charleston	Recip. oil fuel	1	11	23.75	31.70	....	....
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	18.00	36.00	....	....
	Freight Lighter	Navy Yard, Portsmouth	.....	....	....	92 00	95.00	....	....
	Water Barge	Navy Yard, Mare Isl d	.....	....	....	99.00	99.20	....	....
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	88.6	88.4
40	*L-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.9	98.9
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.5	98.5
42	*L-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96 89	96.89	98.1	98.1
43	*L-4*	Elec. Boat Co., Quincy.	Diesel-New Lond	2	14	95.99	95.99	98.1	98.1
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	47.40	52.30	88.5	80 0
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	34.84	39.21	76.8	73.5
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	30.20	36.67	74.8	71.5
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96.04	97.18	94.6	94 5
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	8,51	11.61	72.4	72 0
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	94.84	94.90	91.3	90.5
50	*L-10*	Elec Boat Co., Quincy	Diesel-New Lond	2	14	94.07	94.57	90.6	89.5
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	93.02	93.52	87.7	85.5
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	....	....	....	....
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	33 32	42.21	48.4	41.2
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	33.32	42.27	48.4	41.2
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	31.73	34.60	48.4	41.2
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	8.21	8.31	55.0	49.2
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	8.21	8.31	53.2	48.1
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	8.21	8.31	52.0	46.8
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	8.21	8.31	53.1	47.9
62	*O-1*	Navy Yard, Portsmouth	Diesel-New Lond	2	14	....	....	....	....
63	*O-2*	Navy Yard, P. S.	Diesel-New Lond	2	14	....	....	....	....
64	*O-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
65	*O-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
66	*O-5*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
67	*O-6*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
68	*O-7*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
69	*O-8*	Elec Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
70	*O-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
71	*O-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	....	....	....	....
72	*O-11*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	....	0.07	1.8	....
73	*O-12*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	....	0.07	1.9	....
74	*O-13*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	....	0.07	1.9	....
75	*O-14*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	....	0.07	....	....
76	*O-15*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	....	0.07	....	....
77	*O-16*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	....	0.07	....	....

SUBMARINES FOR JAPAN.

It is stated in the "Japan Advertiser" that in the Kawasaki Dockyards Company of Kobe, has entered into a contract with the Fiat San Giorgio Company of Turin for the construction of the latest model submarine of the *Laurente* type This boat displaces 700 tons when on the surface and 1,070 tons when submerged. In speed it is as fast as any modern battleship and can travel over 6,000 miles without replenishing its supply of oil. The length is 75 meters. It carries five torpedo tubes, three in the bow and two in the stern, besides anti-aircraft submersible guns and other up-to-date appliances. The hull of this submarine will be built in Japan at the Kawasaki Dockyards. The machinery and engines, etc., are to be built and sent from Italy. The vessel will be completed within 24 months. The Kawasaki Dockyards have obtained all the rights and privileges to build all future engines and machinery if they so desire in Japan. All the plans and details with the rights are to be furnished them. A further contract has been signed which gives the company the exclusive rights to manufacture in Japan the "Fiat" type of Diesel oil engines. These engines are applicable not only to submarines but are now proving successful and economical factors as marine engines even in larger vessels. For the construction of the hull of the first submarine and for instructions as to operating, several Italian engineers will go to Japan. The Kawasaki Dockyards Company will send a number of technical experts to study in the Fiat works in Turin and in Spezia.—"Page's Engineering Weekly."

BRISBANE—FIRST AUSTRALIAN-BUILT CRUISER.

The launching of the cruiser *Brisbane* in Sydney recently, marked an epoch in the history of the Commonwealth.

Not only was the *Brisbane* the first cruiser to be built by any of the great dependencies of the Empire, but she was the largest ship of any type yet sent afloat in the southern hemisphere of Greater Britain. Whilst the period occupied in the construction necessarily exceeded that of British yards, it must not be forgotten that a staff had to be created and much work of a novel character taken in hand by Australian artisans. The boilers, for instance, have been entirely built at Cockatoo, and while the material of the main propelling machinery has been imported the turbines themselves have all been assembled and put together by the engineering staff of the island.

The launching ways were 343 feet long, each side being 2 feet 9 inches wide. Before reaching the water the *Brisbane* had to travel 365 feet down the ways before her stern became water-borne, and nearly 500 feet before she was entirely afloat. While moving down the massive concrete ways the load carried by each square foot of concrete was estimated to be approximately two tons, and in order to facilitate the movement no less than 3½ tons of tallow were utilized in greasing the ways. Owing to the fact that the rise and fall of the tide in Sydney was only some 6 feet, it became necessary to provide some means of greasing the ways, which at high water are, of course, submerged. The difference between high and low water would not have been sufficient to ensure the steady progress of the ship into her element, and consequently it became imperative to provide some method of keeping the water back whilst the ways were being treated. In order to do this a coffer dam was temporarily built, and as soon as the water was pumped out, what is in effect a small dock with a floating caisson was built. This is, of course, a

permanent work, and in time to come will be available for vessels of even greater tonnage than the *Brisbane*.

Originally the work of building the *Brisbane* was entrusted by the Commonwealth Government to the State authorities, but it was found in practice that the dual control was cumbersome, and the Commonwealth acquired the Cockatoo establishment and undertook its own work. Whilst the ship was being framed the superintendence of the work was entrusted to Mr. J. W. Clark, M. I. N. A., an Admiralty official, who is now assistant general manager of the yard. The Cook Government having decided to import a general manager of the highest qualification, applied to the Admiralty for the services of a member of the Royal Corps of Naval Constructors, a body of highly-skilled officials, whose reputation is world-wide. In consequence Mr. Julian King-Salter was selected for the appointment, and since his arrival in Sydney the establishment has gradually assumed the system and methods of the great naval yards of Great Britain. With two acknowledgedly skilled officials at the head of affairs, it may confidently be expected that ships launched from Cockatoo will bear more than favorable comparison with those of Europe.—"Page's Engineering Weekly."

BOOKS RECEIVED.

WIRELESS TELEGRAPHY, by DR. J. ZENNECK.—Seelig's Translation.

Prof. Zenneck's "Wireless Telegraphy" is the most complete modern and comprehensive book on the subject yet published. A study of this volume brings the reader up to date in radio, technically. The author has delved into details sufficiently to interest the learned, yet at the same time his explanations are in language simple enough to hold the attention of the less skilled. The chapters concerning undamped wave machines and arcs are of especial interest. The esteem in which Prof. Zenneck is held in the scientific world adds to the value of the publication.

ASSOCIATION NOTES.

A meeting of the Society was called on December 30, 1915, at which time the ballots cast for the officers of the Society for the year 1916, and for and against the holding of a banquet in that year, were counted.

The following officers were elected:

President, Captain C. W. Dyson, U. S. Navy.
Secretary-Treasurer, Lieutenant A. T. Curch, U. S. Navy.

Council:

Captain T. W. Kinkaid, U. S. Navy.
Engineer-in-Chief C. A. McAllister, U. S. Coast Guard.
Lieutenant Commander J. O. Richardson, U. S. Navy.

The vote on the banquet resulted as follows:

For, ...	227
Against, ...	84

The banquet of the Society for 1916 will be held on Saturday, March 25, 1916, at 7:30 P. M., at the Army and Navy Club, Washington, D. C.

The committee having charge of arrangements for the banquet is as follows:

Lieutenant Commander H. C. Dinger, U. S. Navy;
Lieutenant Commander J. H. Tomb, U. S. Navy;
Lieutenant Commander J. O. Richardson, U. S. Navy.

The following members and associates have joined the Society since the publication of the last number of the JOURNAL:

MEMBERS.

Allen, Henry A., Consulting Engineer, 913 Galt Avenue, Chicago, Ill.
Armstrong, Daniel W., Ensign, U. S. Navy.
Bowen, D. F. X., Captain of Engineers, U. S. C. G.
Chambers, Irving R., Ensign, U. S. Navy.
Cobb, Calvin H., Lieutenant, U. S. Navy.
Coyle, John B., 1st Lieutenant of Engineers, U. S. C. G.
Edwards, Walter A., Lieutenant, U. S. Navy.
Elmer, Robert E. P., Ensign, U. S. Navy.
Gillette, Claude S., Lieutenant, U. S. Navy.
Hammond, T. E., The Palace Apartments, Long Beach, Cal.
Little, Harold H., Ensign, U. S. Navy.
McGlasson, Archibald, Lieutenant, U. S. Navy.
McMillin, George J., Lieutenant, U. S. Navy.
Matheis, Aaron, 3d Lieutenant of Engineers, U. S. C. G.
Norton, Homer H., Lieutenant, U. S. Navy.
Palmer, Edward F., 3d Lieutenant of Engineers, U. S. C. G.
Redman, Joseph R., Ensign, U. S. Navy.
Riheldaffer, John L., Ensign, U. S. Navy.
Sherman, Frederick C., Lieutenant, U. S. Navy.
Smith, Karl F., Lieutenant, U. S. Navy.
Stoer, C. H., 603 Finance Building, Philadelphia, Pa.
Swift, William, Rear Admiral, U. S. Navy (Retired).
Woodruff, George L., Lieutenant, U. S. Navy.
Woodside, Elmer L., Ensign, U. S. Navy.

ASSOCIATES.

Adams, Harold M., Bureau of Steam Engineering Navy Department, Washington, D. C.
Fernald, Benjamin G., Vice President and District Manager, Kerr Turbine Co., 95 Liberty St., New York City.
Fryer, Ross Lauder, 409 Riggs Building, Washington, D. C.
Hardesty, Fred S., Consulting Engineer, Riggs Building, Washington, D. C.

King, Joseph B., Midvale Steel Co., Philadelphia, Pa.

Kothny, G. L. E., M. E., care of C. H. Wheeler Mfg. Co., 18th St. and Lehigh Avenue, Philadelphia, Pa.

Newton, N. A., Sales Manager and Chief Engineer, National Transit Co., Oil City, Pa.

O'Neill, J. G., Chemist, Naval Engineering Experiment Station, Annapolis, Md.

Porter, J. Benton, General Electric Co., Witherspoon Building, Philadelphia, Pa.

Reid, John A., N. A., John Reid & Co., 17 Battery Place, New York City.

Risberg, Gustaf, Constructor, R. Swedish Navy, Hotel Schenley, Pittsburgh, Pa.

Schombert, John H., Bureau of Steam Engineering, Navy Department, Washington, D. C.

Stuart, M. C., M. E., Naval Engineering Experiment Station, Annapolis, Md.

Swygert, George H., Bureau of Steam Engineering, Navy Department, Washington, D. C.

Thurston, Perry K., Bureau of Steam Engineering, Navy Department, Washington, D. C.

Visintini, Marcello, 221 33d St., Newport News, Va.

Walton, George E., Colorado Building, Washington, D. C.

White, C. E. M., Navy Building, Lousdale St., Melbourne, Australia.

FINANCIAL STATEMENT, 1915.

The following statement shows the returns from and costs of the issue of the JOURNAL of the Society for the year 1915:

RETURNS.

Dues	$5,171.30
Subscriptions	1,714.37
Sales	957.80
Advertsements	2,100.00
Exchange	.78
	$9,944.25

COSTS.

Printing	$4,584.42	
Engraving	599.50	
Drafting	89.45	
Manuscript	1,250.15	
Postage and expressage	152.70	
Commission	133.34	$6,809.56
Profit on issue of JOURNAL		$3,134.69

GENERAL PROFIT AND LOSS STATEMENT.

The following statement shows the results of the business as a whole during the year 1915:

	Loss.	*Profit.*
Profit on issue of JOURNAL		$3,134.69
Interest		339.91
		$3,474.60
Banquet, 1915	$262.59	
General expense (includes salaries)	1,618.61	
Current profit and loss account	27.05	1,908.25
Net profit		$1,566.35

BALANCE STATEMENT.

The following statement shows the difference between the total assets on January 1, 1915, and those on December 31, 1915:

Assets December 31, 1915.

Cash	$2,607.14
Accounts receivable, advertisements	1,405.11
Accounts receivable, dues	807.50
Accounts receivable, sales	195.33
Accounts receivable, subscriptions	426 06
Army and Navy Club bond (1) (cost)	1,000.00
Washington Railway and Electric bonds (6) (cost)	5,215.50
Furniture	62.50
Miscellaneous asset (gold medal)	16.00
	$11,735.14
Total assets January 1, 1915	10,168.79
Net gain	$1,566.35

Market value of Washington Railway and Electric bonds carried above at cost was, on December 31, 1915, 81.

Army and Navy Club bond is not listed.

Audited and found correct.

The numerical strength of the Society on December 31, 1915, was—

Active members,	702
Associate members,	361
Subscribers,	346
Exchanges,	77
Total,	1,486

Audited and found correct.

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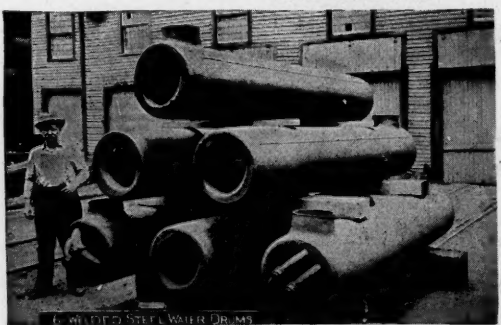

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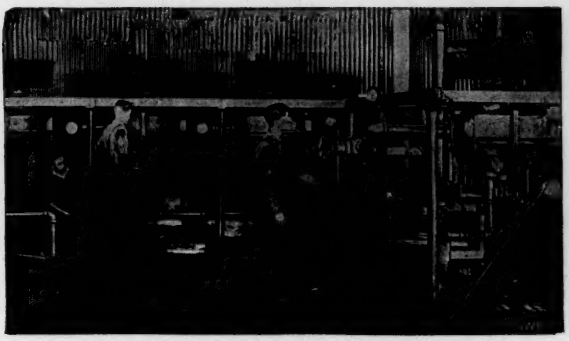

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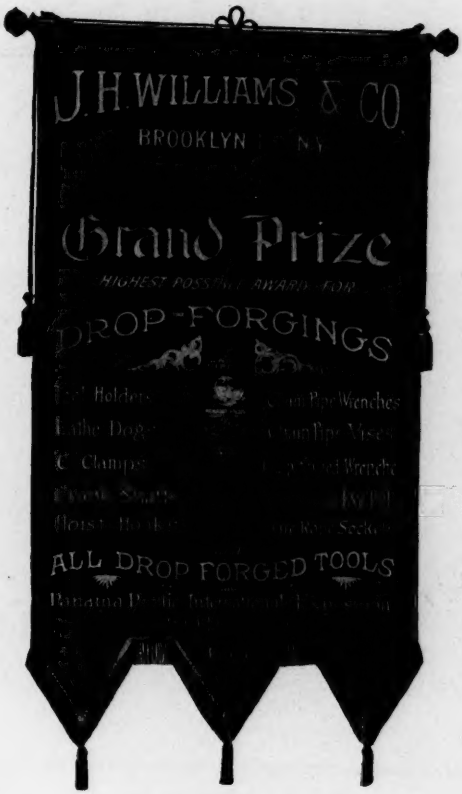

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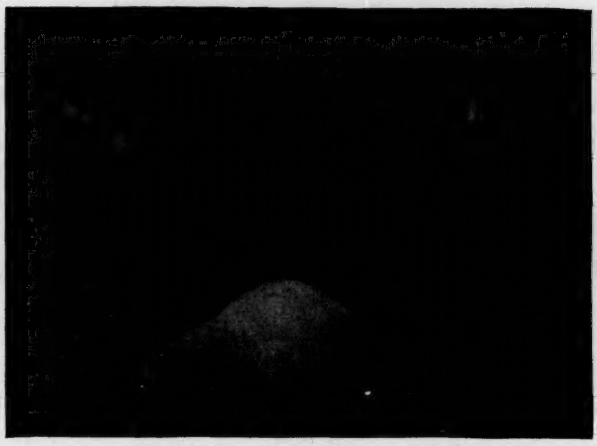

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The Lovekin Assistant Cylinders are designed for balancing the forces due to **BOTH THE INERTIA AND WEIGHT OF THE VALVE GEAR.** They are, therefore, of great value to engines **FITTED WITH EITHER PISTON OR FLAT SLIDE VALVES.**

Several of the United States Naval Vessels in commission and all those now under construction have their engines fitted with LOVEKIN PATENT ASSISTANT CYLINDERS **TO ALL OF THEIR VALVE GEARS.**

The following Shipbuilding and Engineering Companies have fitted LOVEKIN ASSISTANT CYLINDERS to valve gears on engines built by them:

New York Shipbuilding Co.	Camden, N. J.
Newport News Shipbuilding Co.	Newport News, Va.
Wm. Cramp & Sons Shipbuilding Co.	Philadelphia, Pa.
Fore River Shipbuilding Co.	Quincy, Mass.
American Shipbuilding Co.	Cleveland, Ohio.
Vulcan Works	Stettin, Germany.
Kawasaki Dockyard Co.	Kobe, Japan.
Societa Anonima	Milan, Italy.
N'Odero Co.	Genoa, Italy.

Many of these vessels have run over 250,000 miles without requiring any overhauling of the valve gear. These Assistant Cylinders prevent excessive wear on the eccentric straps or any parts of the valve gear **AND THEREFORE ENABLE THE VALVE GEAR TO BE KEPT IN CORRECT ADJUSTMENT FOR VERY LONG PERIODS SO THAT THE STEAM DISTRIBUTION THROUGH THE VALVES AND PORTS REMAIN AS DESIGNED.**

It is surprising to see the great consideration that is given the flat slide valve frequently used on L.P. Cylinders, few engineers forget to state that "balance cylinders will be used for the L.P. Valve Gears." How many engineers or designers have taken the trouble to find out what a poor device the ordinary balance cylinder is? It can only hold the weight of the valve gear up at one particular receiver pressure. The inertia force which is ever present and varying with the speed of the engine is always the force that causes the trouble and requires a continual expense for upkeep. **THESE INERTIA FORCES ARE PRESENT TO THE SAME EXTENT IN ALL VALVE GEARS IF THEY ARE OF THE SAME WEIGHT,** therefore, why specify the L.P. Valve Gears to be balanced and let the H.P. and I.P. Valve Gears go unbalanced when all valve gears can be balanced so easily by the LOVEKIN ASSISTANT CYLINDER.

THE LOVEKIN ASSISTANT CYLINDER performs the work of neutralizing the forces set up in the valve gear, **DUE TO BOTH THE INERTIA AND WEIGHT** of the same, and, therefore, **SAVE POWER, SAVE WEAR AND TEAR, SAVE OIL, SAVE REPAIR BILLS AND KEEP THE VALVE SETTING CORRECT FOR MUCH LONGER PERIODS THAN IS POSSIBLE WITHOUT THEM.**

The valve gear can be designed much lighter.

Balancing the inertia and gravity forces in the valve gears enables the main engine to be balanced to a far greater extent than is possible where the valve gear forces are not balanced. This was proven quite recently in the case of some high-speed destroyers in the United States Navy, where comparative tests were made with and without LOVEKIN INERTIA CYLINDERS.

NOTE—NO FORM OF PISTON VALVE OR ORDINARY BALANCE CYLINDER CAN BE DESIGNED TO DO MORE THAN CARRY THE WEIGHT OF THE VALVE GEAR AT ONE CONSTANT PRESSURE IN THE RECEIVER. THEREFORE THE GREATEST FORCE, OR THAT DUE TO THE INERTIA OF THE VALVE GEAR, IS NEGLECTED.

Where flat slide valves are used it is necessary to provide a relief frame at the back of the valve so as to eliminate friction. I have about one thousand of these Cylinders in use at the present time, all of which are giving the greatest possible satisfaction to the owners.

I refer you to the following owners and builders, some of whom have had years of experience with my Cylinders:

American-Hawaiian Steamship Company, New York City, N. Y.	(2 Engines, 3,600 I.H.P.)
United States Navy Department, Washington, D. C.	(20 Engines, 200,000 I.H.P.)
Pacific Mail Steamship Company, California	(8 Engines, 30,000 I.H.P.)
Pacific Coast Steamship Company, Seattle, Washington	(5 Engines, 16,500 I.H.P.)
Merchants & Miners Transportation Co., Baltimore, Md.	(3 Engines, 9,600 I.H.P.)
Gulf Refining Company, New York City, N. Y.	(4 Engines, 12,000 I.H.P.)
Texas Oil Company, New York City, N. Y.	(2 Engines, 5,000 I.H.P.)
Standard Oil Company, New York City, N. Y.	(4 Engines, 7,200 I.H.P.)
Old Dominion Steamship Co., New York City, N. Y.	(1 Engine, 1,800 I.H.P.)
Coastwise Transportation Co., Boston, Mass.	(5 Engines, 9,000 I.H.P.)
Mallory Line Steamship Co., New York City, N. Y.	(1 Engine, 7,000 I.H.P.)
American Shipbuilding Co., Cleveland, Ohio	(12 Engines, 20,000 I.H.P.)
Kawasaki Dockyard Co., Kobe, Japan	(2 Engines, 8,000 I.H.P.)
N. Odero Company, Genoa, Italy	(4 Engines, 20,000 I.H.P.)
Vulcan Works, Stettin, Germany	(2 Engines, 5,000 I.H.P.)
United States Revenue Service, Washington, D. C.	(2 Engines, 3,600 I.H.P.)
Chesapeake Bay Line Steamer *Columbia*	(1 Engine, 3,500 I.H.P.)
Miscellaneous Small Engines	(14 Engines, 15,000 I.H.P.)

TOTAL I.H.P. IN USE 376,800.

Many of the above engines have been in use for periods of from five to ten years, and I have numerous letters from the owners testifying to the great advantages of my patent Assistant Cylinders.

MR. LOVEKIN,
Chief Engineer, New York Shipbuilding Co., Camden, N. J.

CAMDEN, N. J., *June 28, 1913.*

DEAR SIR:

I have been Chief Engineer of the S. S. *Governor* for the past 6 years, and your Assistant Cylinders were installed on her. I found them to be a very good thing for the eccentrics, for after running 6 years the low-pressure straps had only worn $\frac{1}{16}$ of an inch, and the H.P. and M.P. the thickness of a tin shim. The engines are twin screw 5,000 I.H.P.

We ran about 370,000 miles and averaged 22 landings a month. We only had water on an eccentric once in that time, and that was when the steam pipe to the low-pressure Assistant Cylinder broke, and while we were repairing it we had to run water on it as it would not run with oil.

The only repairs we had to make to the cylinders was to put in new snap rings once a year, and I can recommend as a great saving on the valve gear and the eccentric straps.

Yours truly,

B. K. MARTLAND,
Chief-Engineer, S. S. *Congress*,
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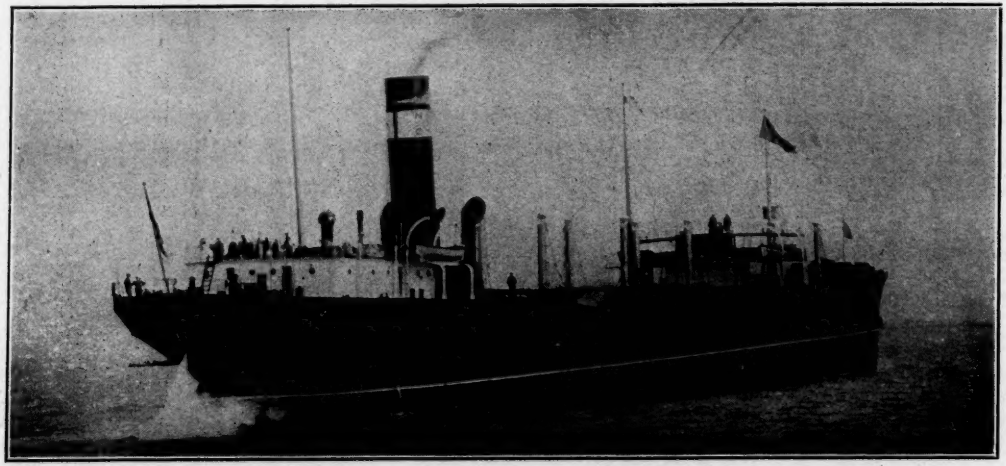

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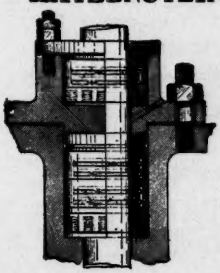

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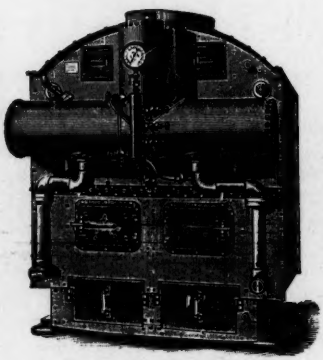

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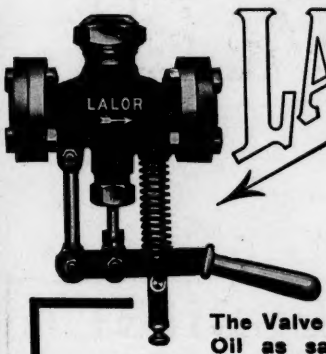

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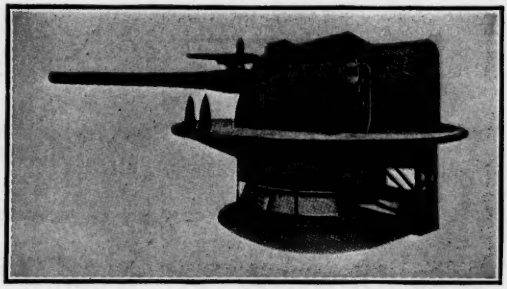

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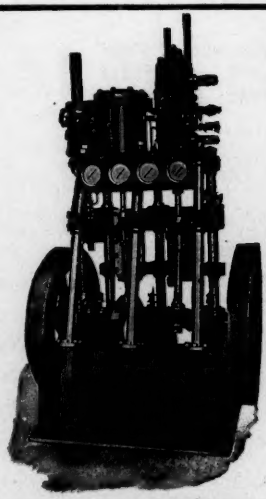

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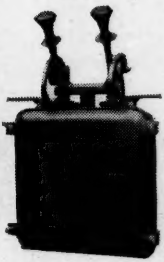

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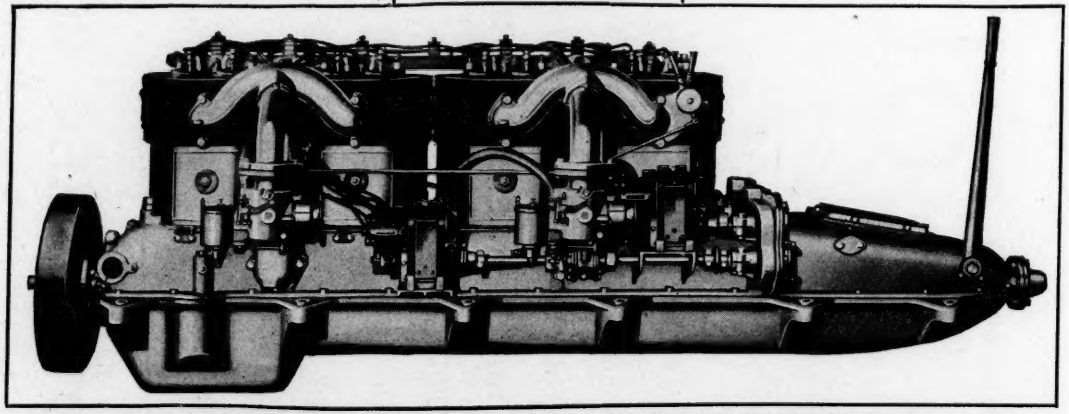

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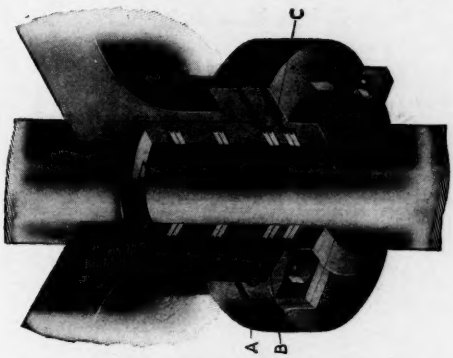

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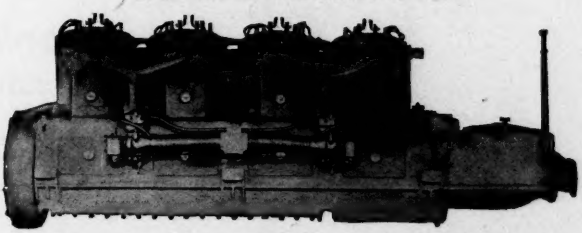

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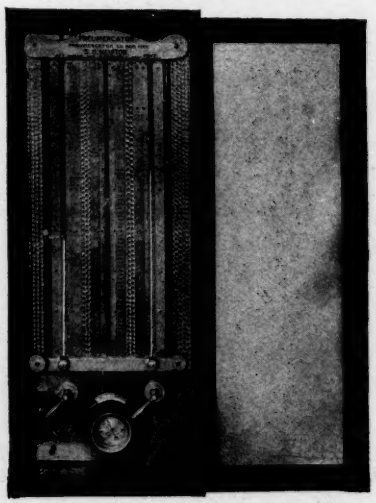

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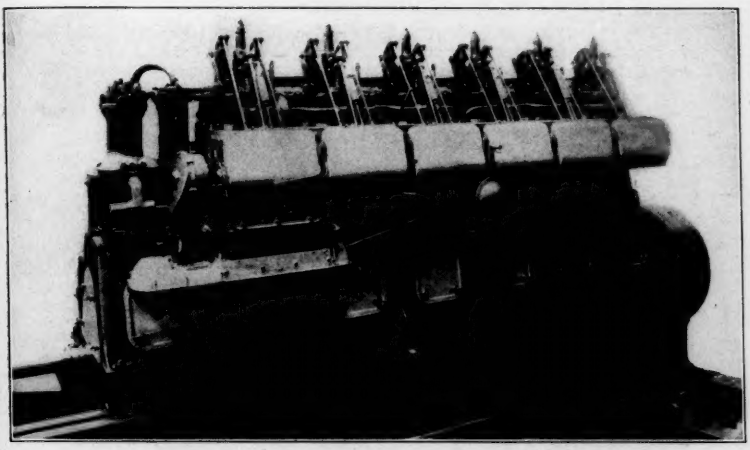

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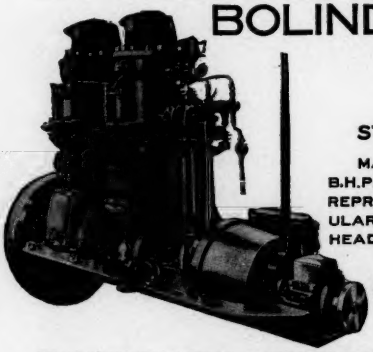

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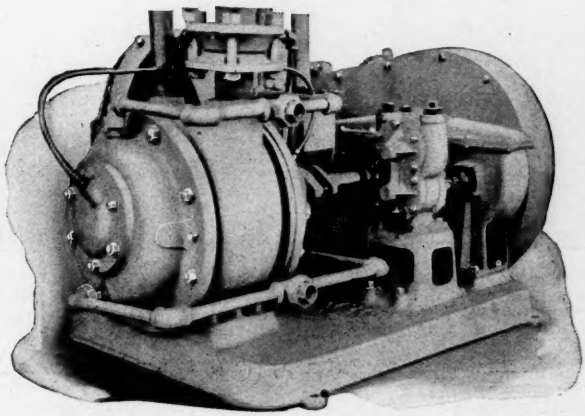

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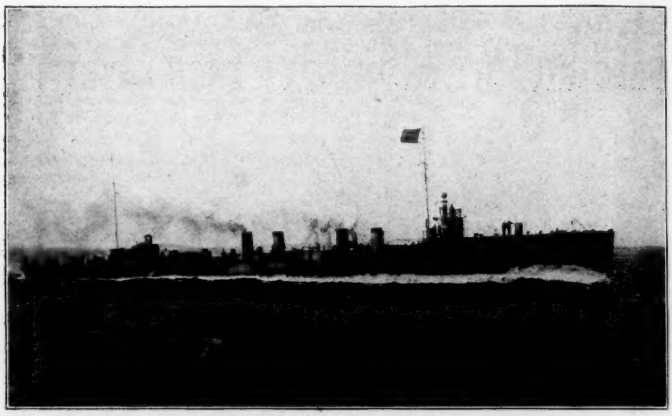

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